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Puiu

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(54) **ACTIVE TORQUE COUPLING WITH HYDRAULICALLY-ACTUATED ROTARY CLUTCH ACTUATOR**

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 272 days.

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(22) Filed: **Jul. 8, 2005**

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(51) **Int. Cl.**

F16D 25/0638 (2006.01)

B60K 17/35 (2006.01)

(52) **U.S. Cl.** **192/35**; 192/85 AA; 192/93 A; 180/249

(58) **Field of Classification Search** 192/35, 192/70.23, 85 AA, 93 A; 180/249, 250; 475/86

See application file for complete search history.

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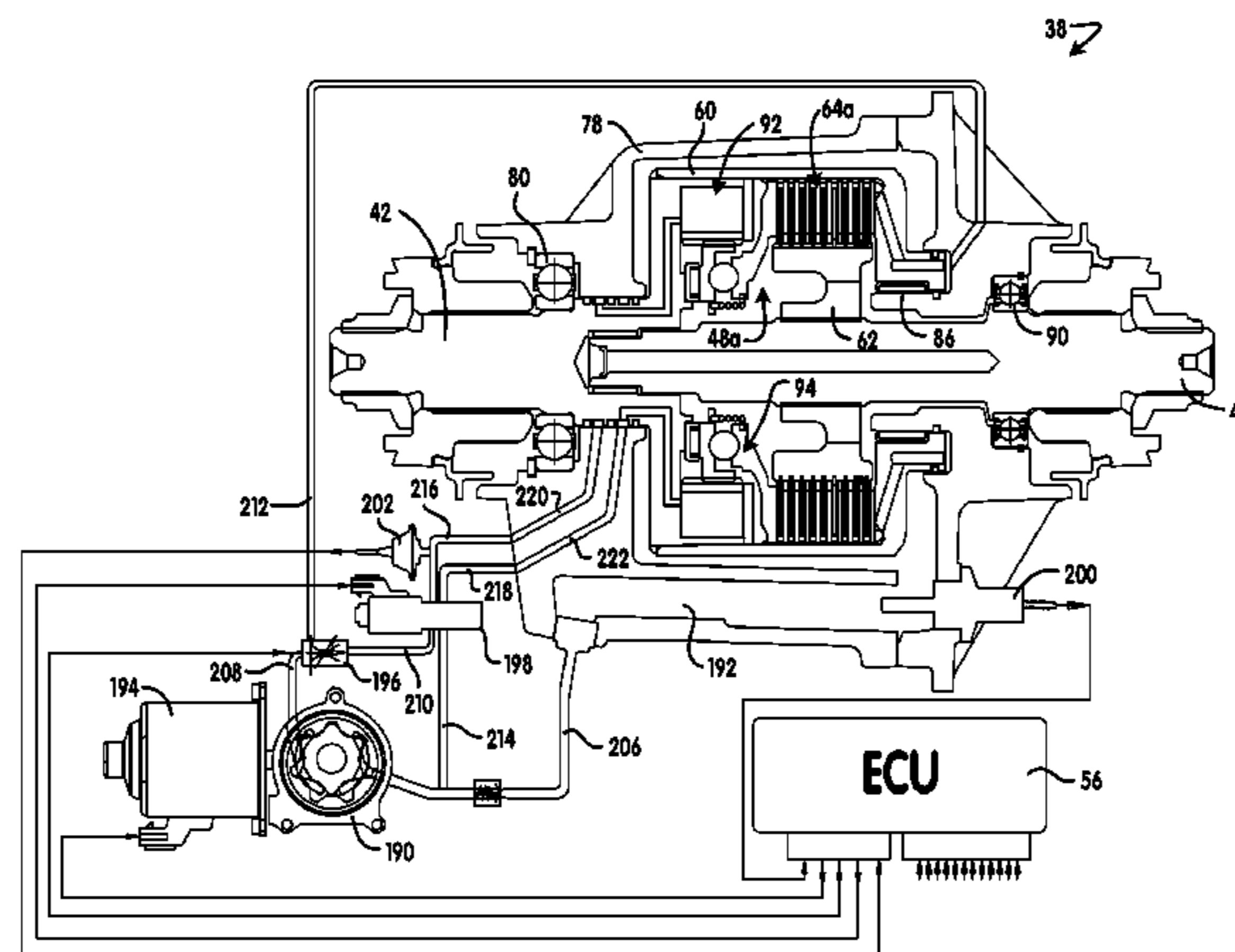
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(57) **ABSTRACT**

A torque transfer mechanism having a multi-plate friction clutch connecting a pair of rotary members and an electro-hydraulic clutch system for controlling engagement of the friction clutch. The electrohydraulic clutch system can include an actuator with first and second components with a plurality of actuation chambers that are employed to control the position of the second component relative to the first component. Relatively high pressure fluid may be directed through or around one or more bypassed portions of the actuation chambers when relatively low compressive clutch engagement forces are to be exerted on the friction clutch. The bypassed portions of the actuation chambers may be sequentially or progressively operated based on a position of the second component relative to the first component to increase an amount of force that is applied by actuator.

36 Claims, 20 Drawing Sheets



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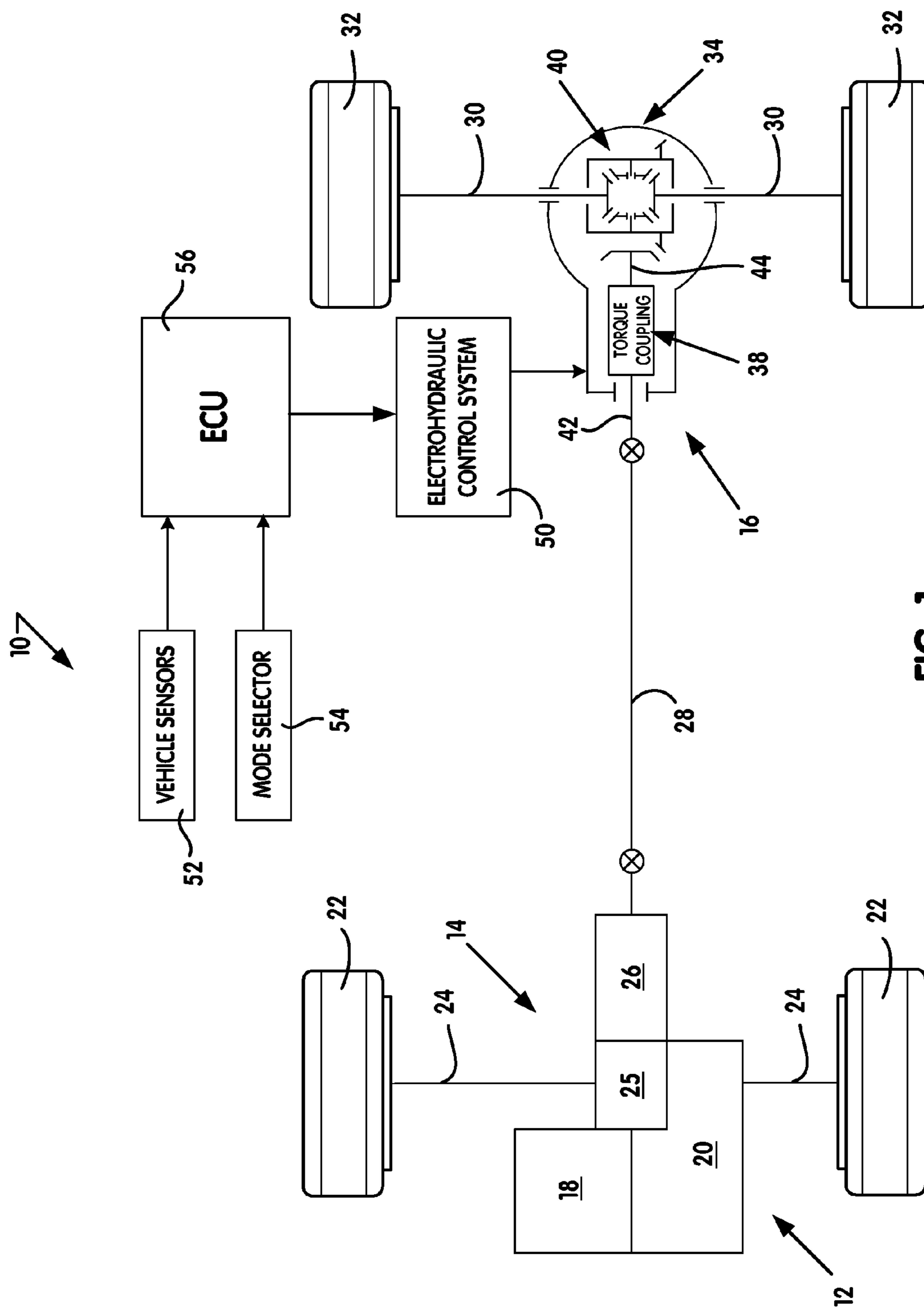
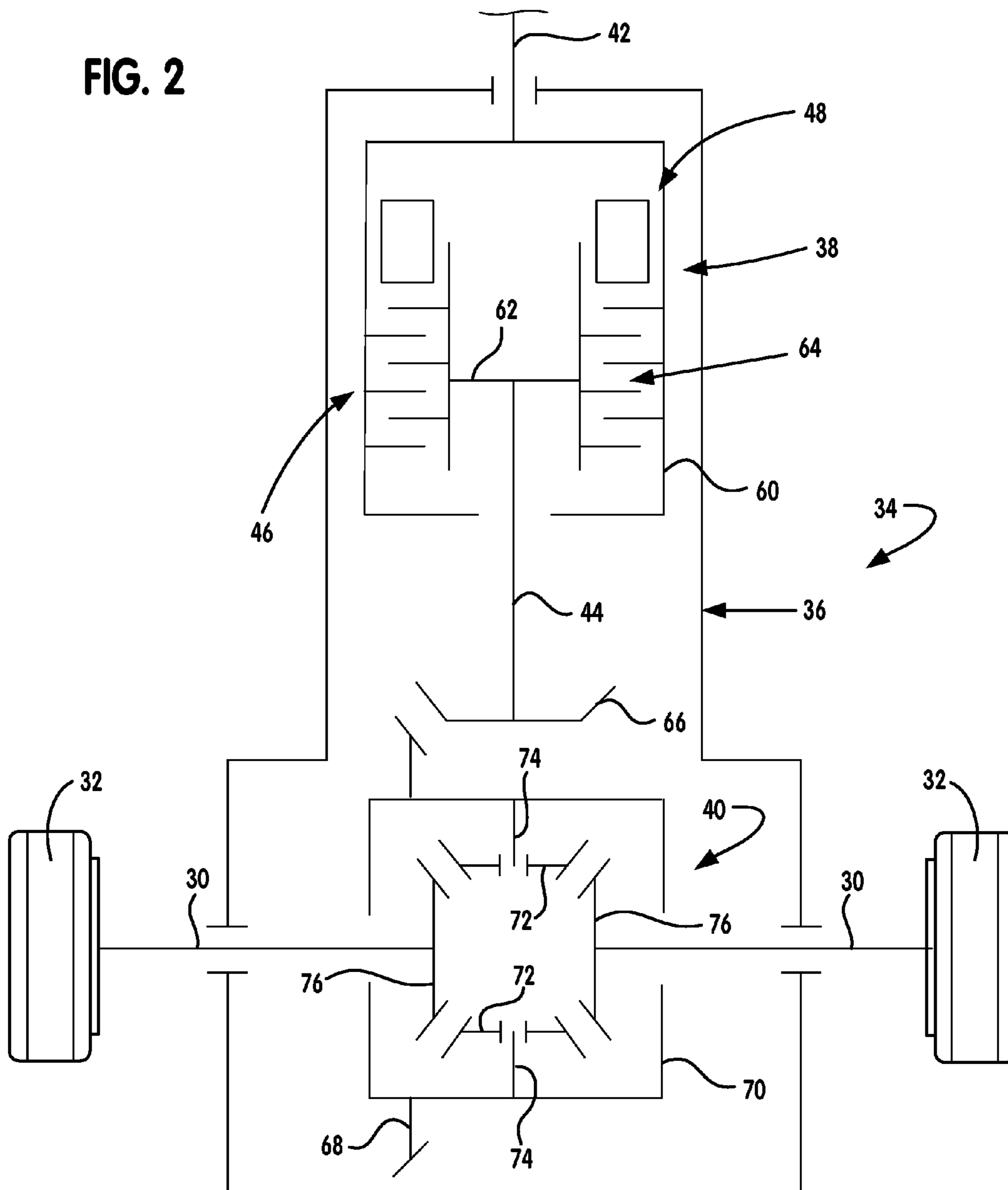


FIG. 1

FIG. 2



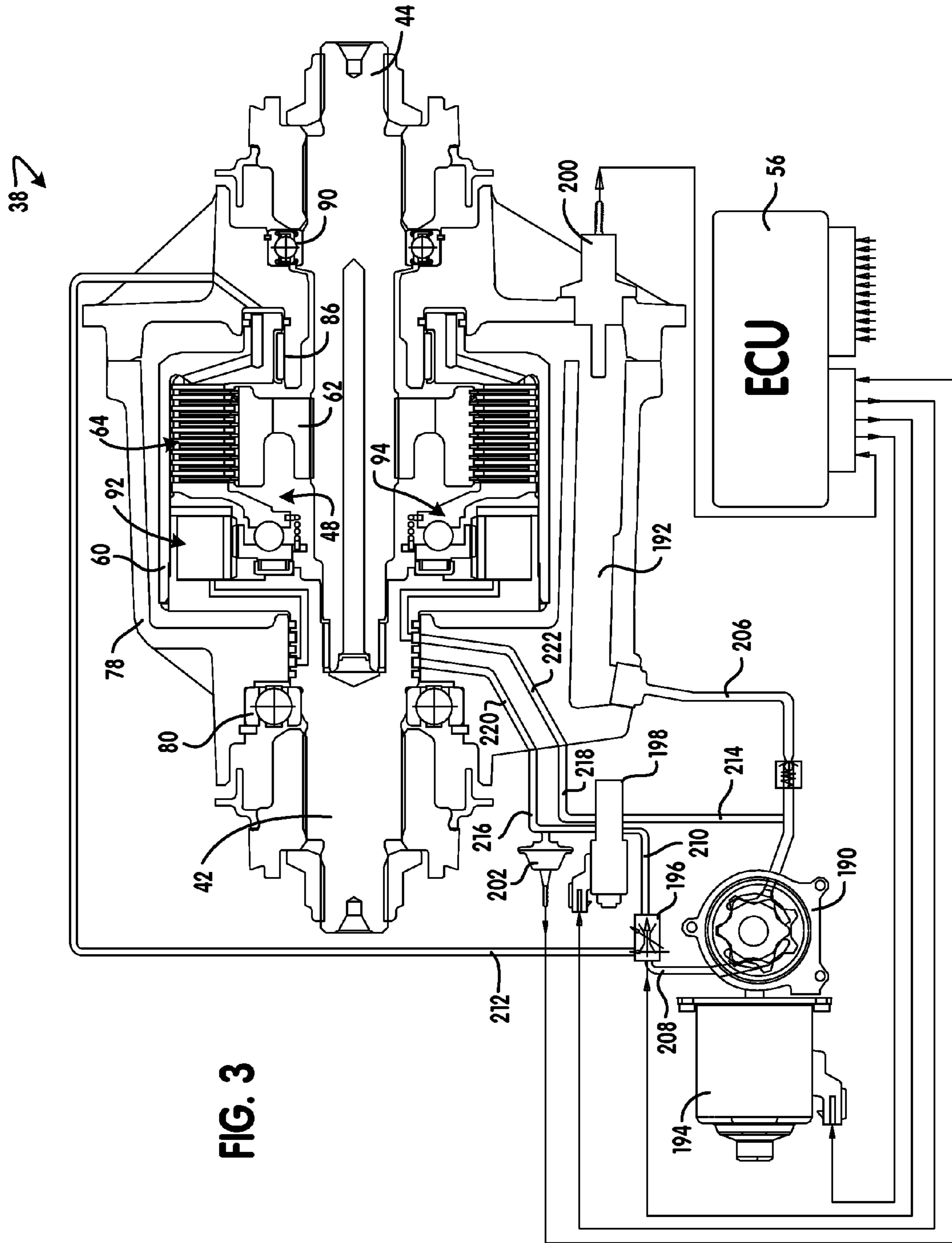


FIG. 3

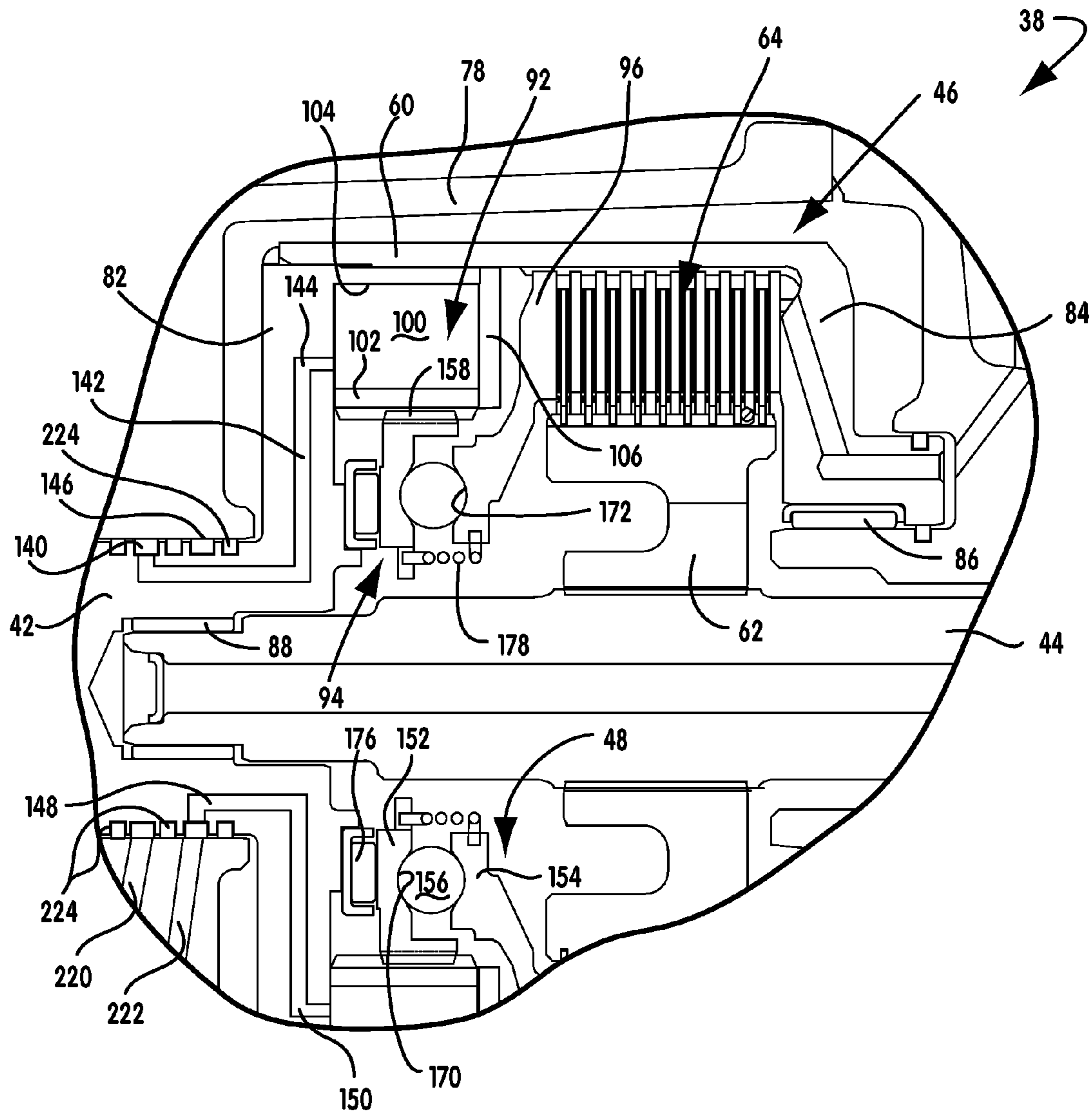


FIG. 3A

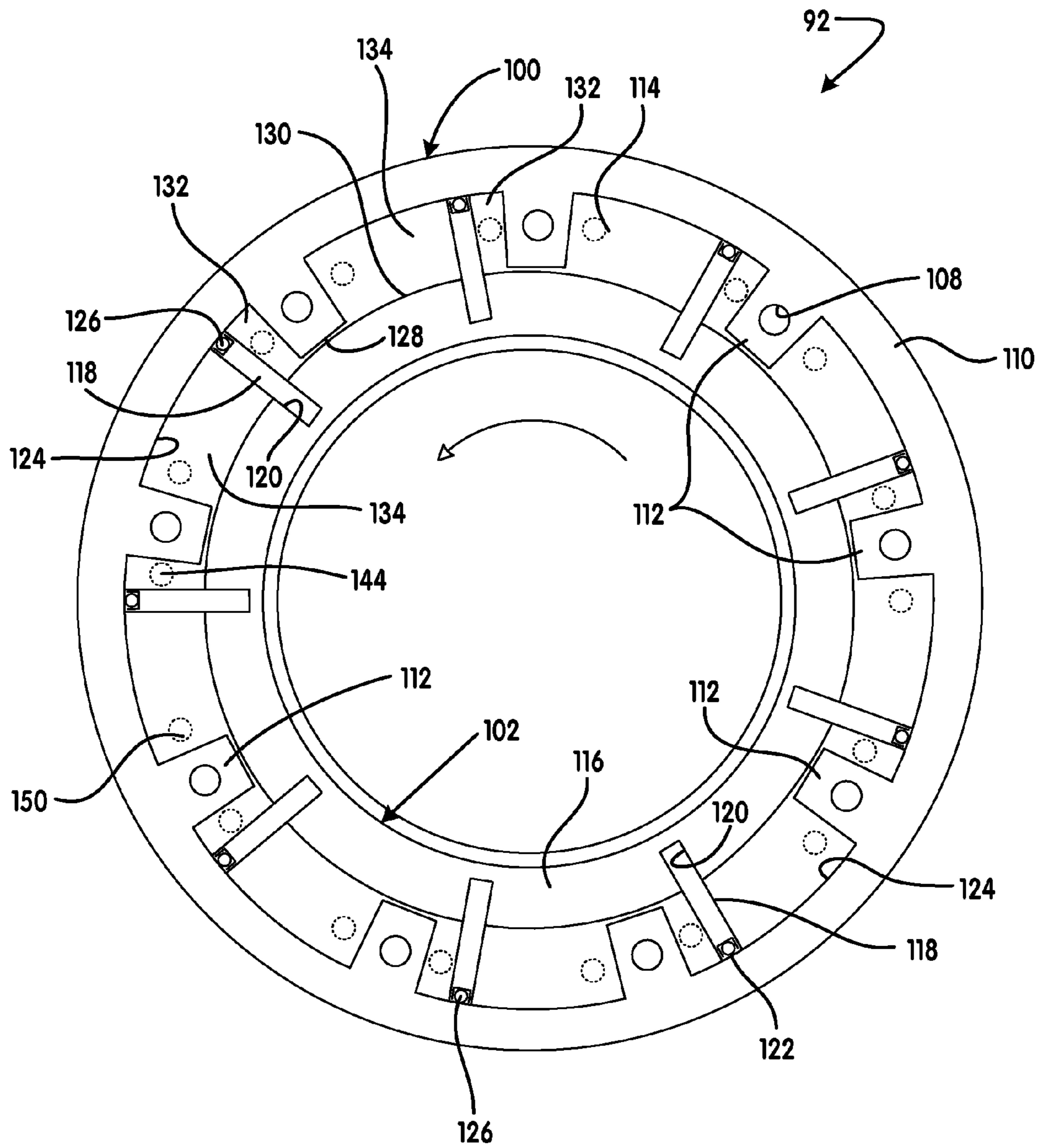


FIG. 4

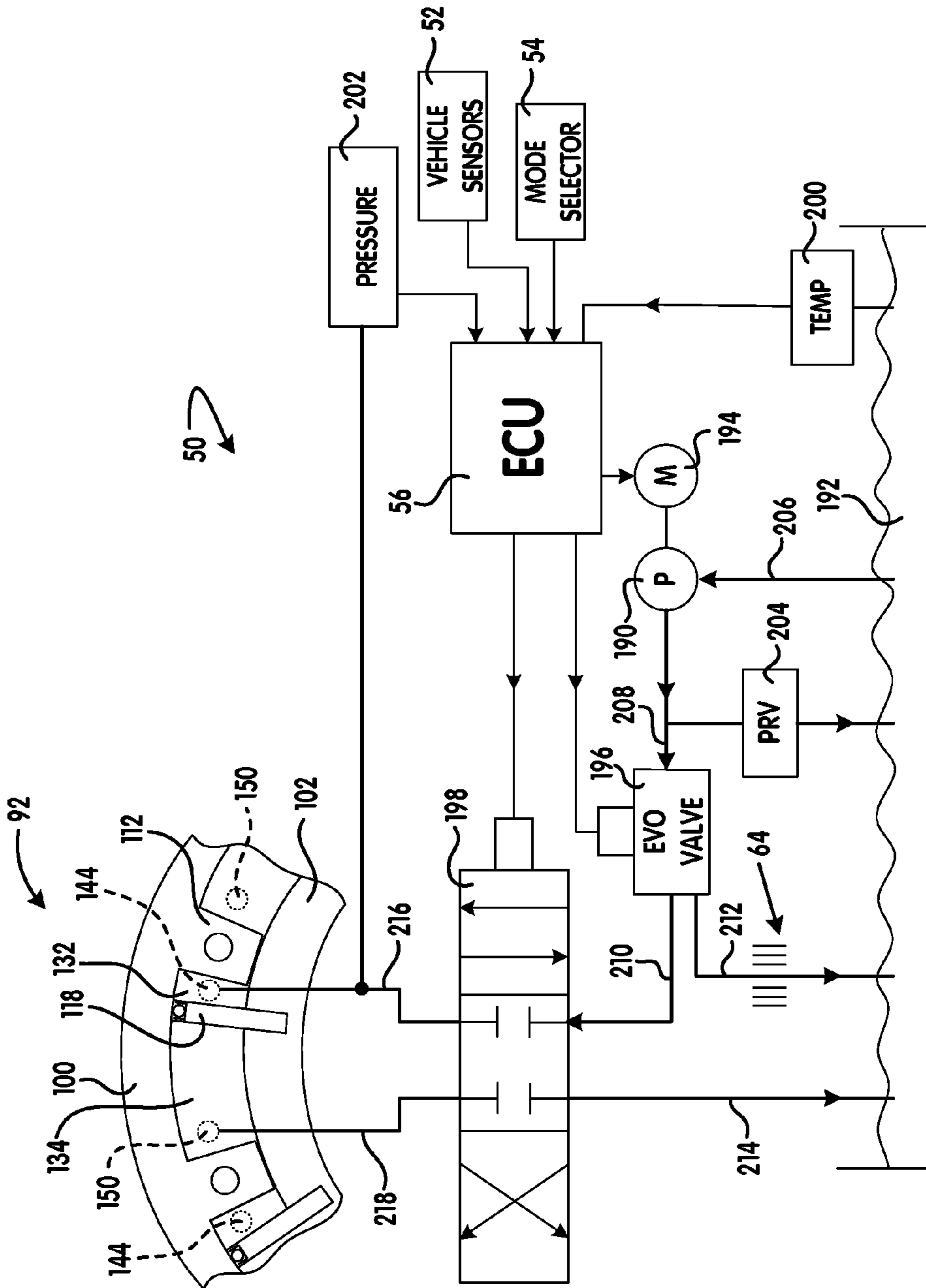


FIG. 5

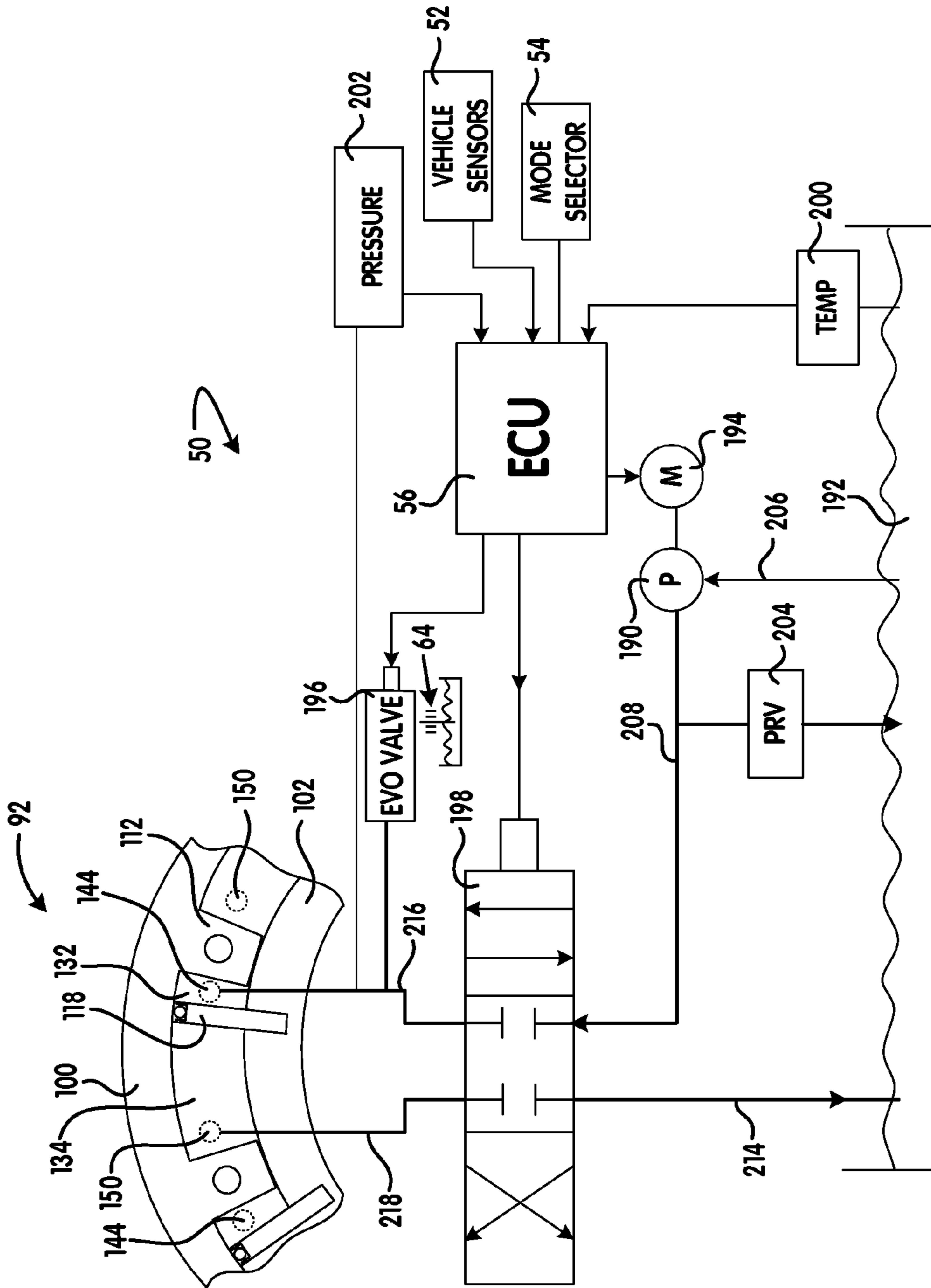


FIG. 6

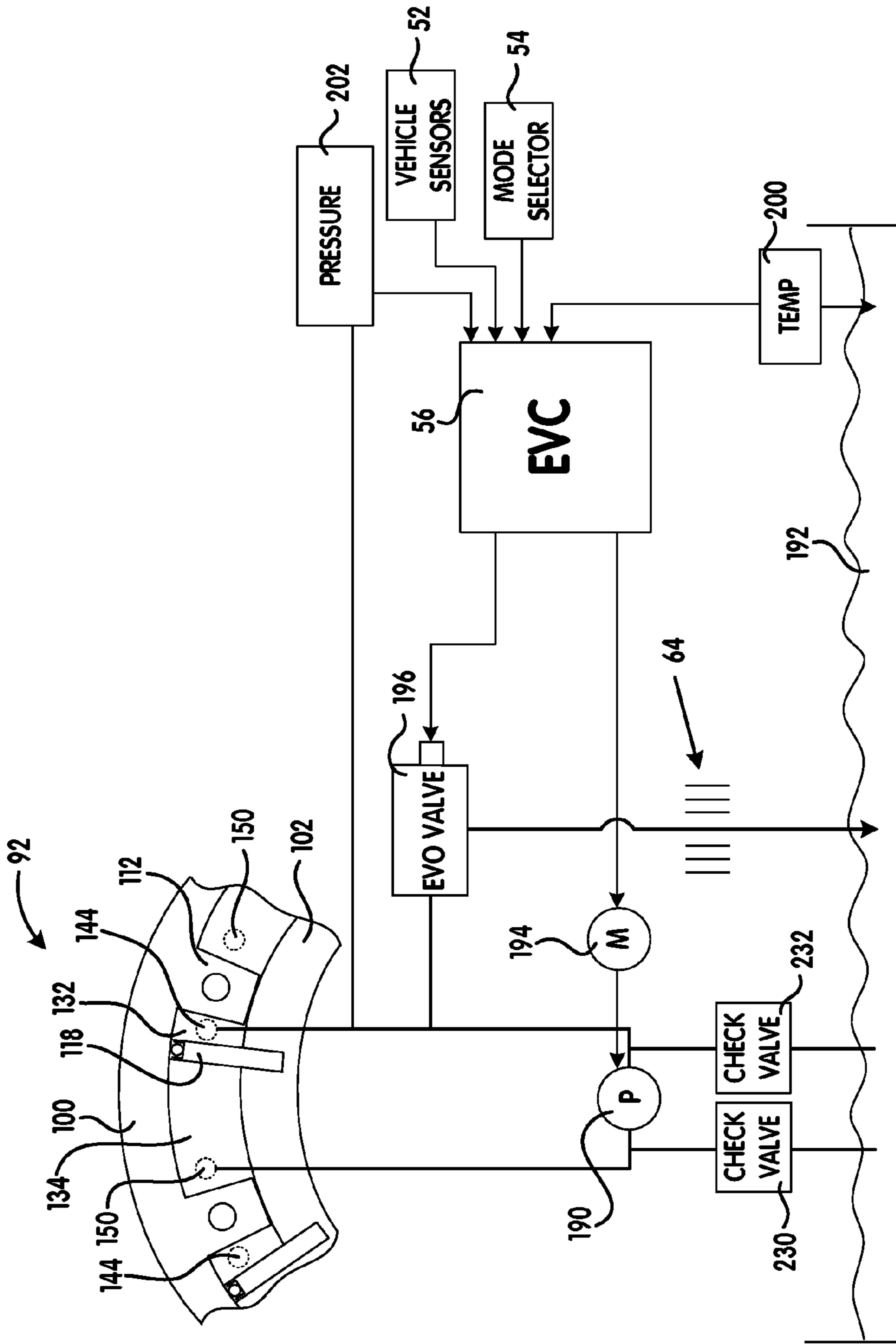


FIG. 7

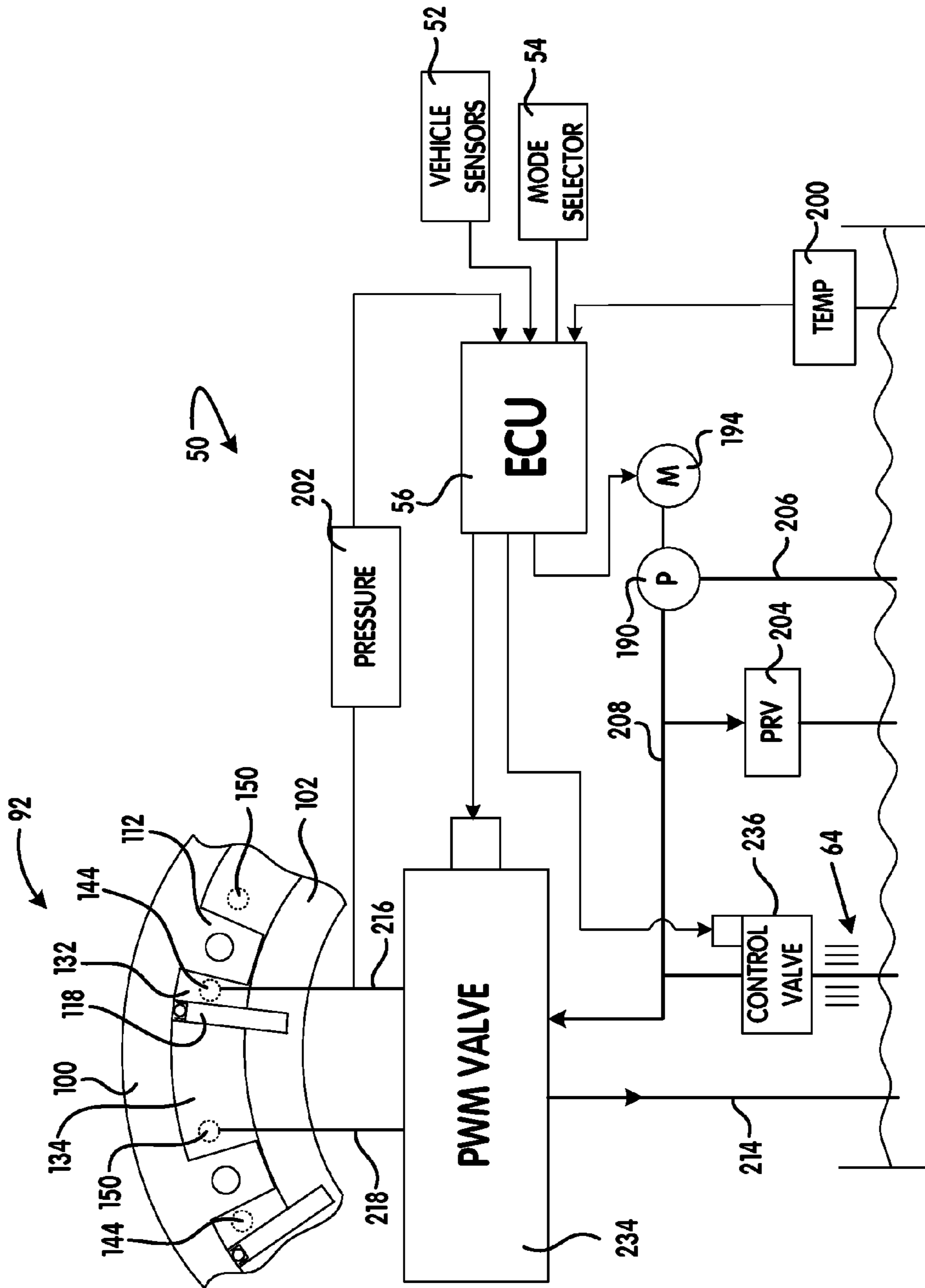


FIG. 8

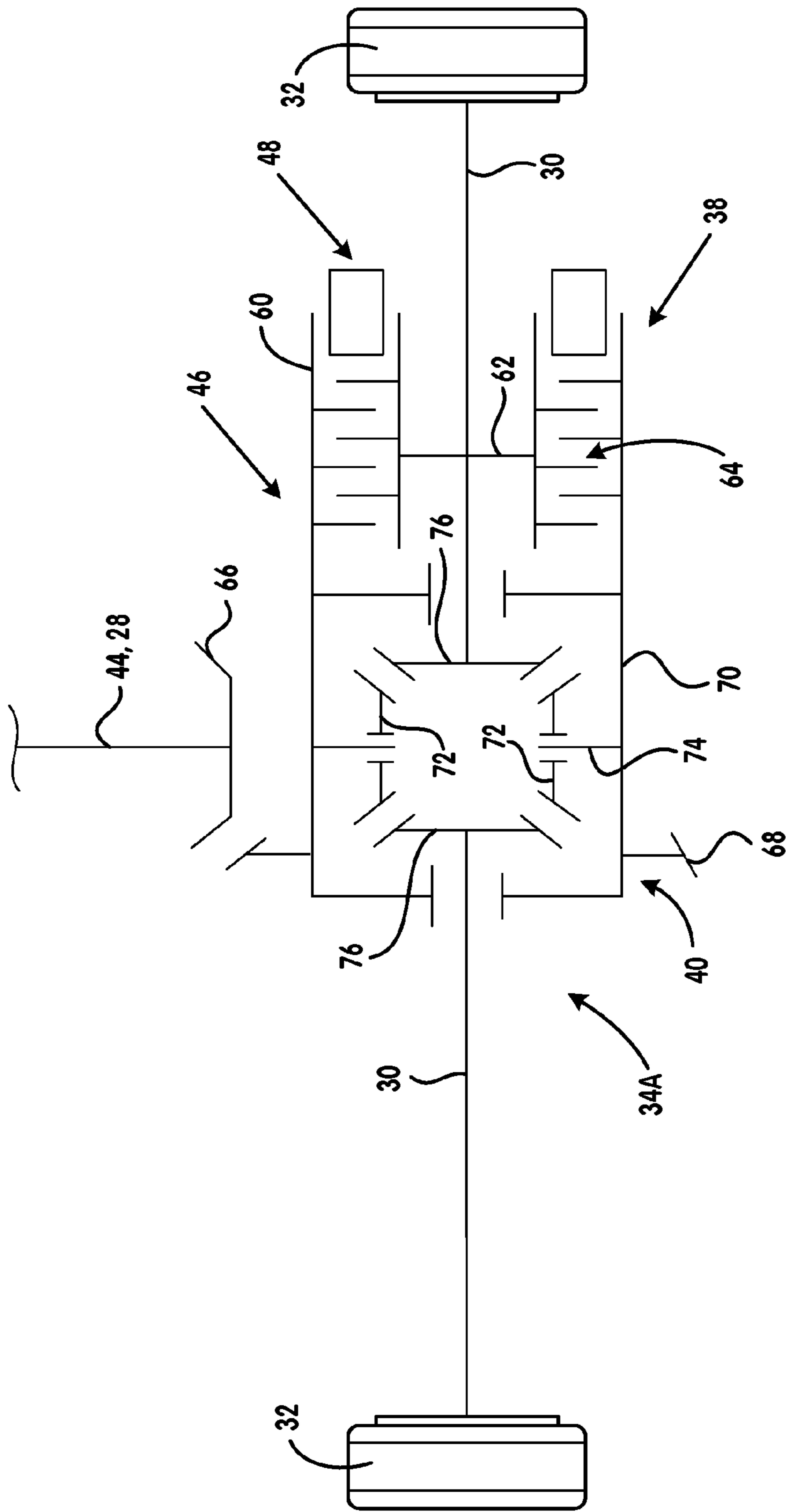


FIG. 9

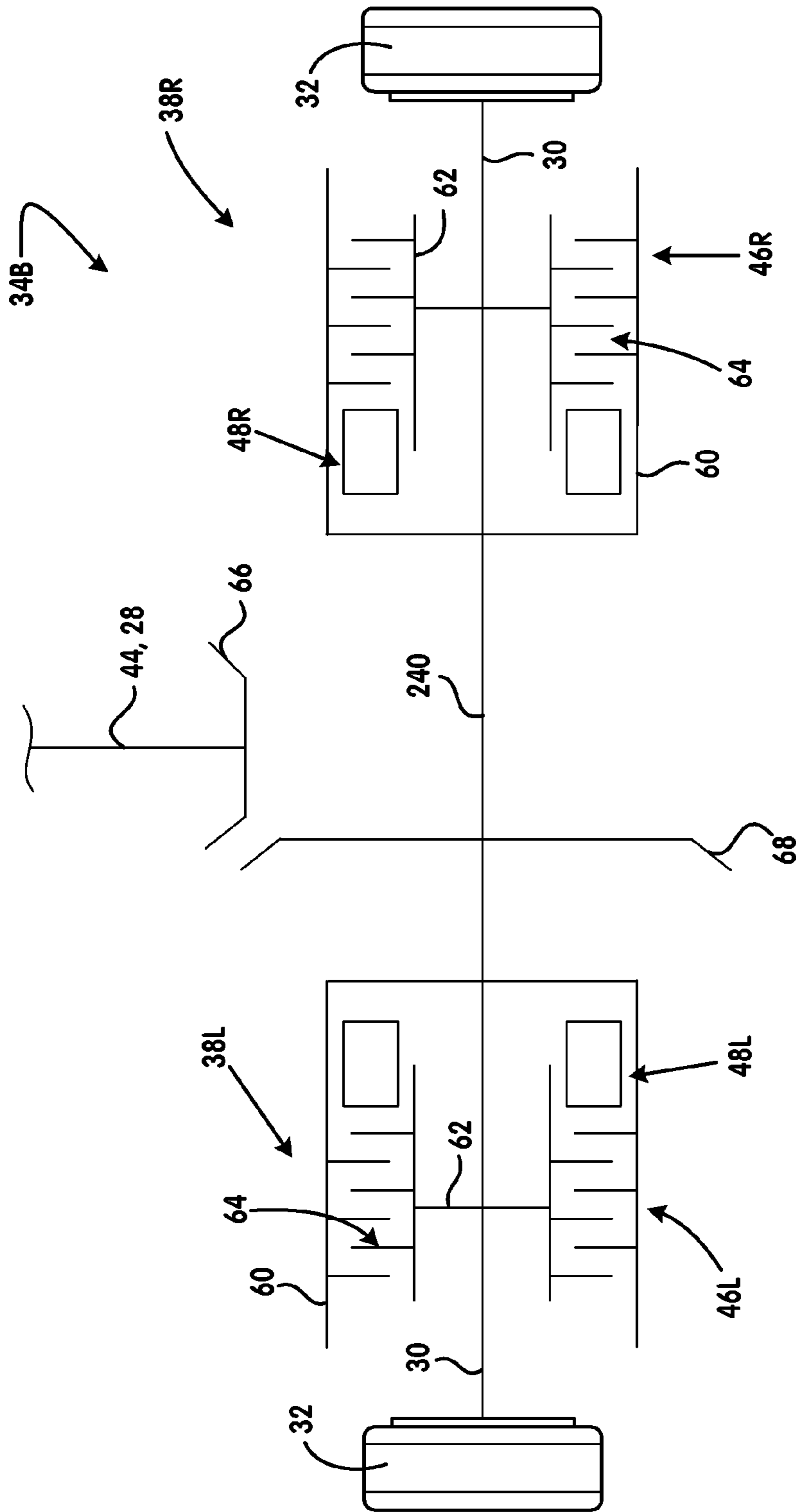


FIG. 10

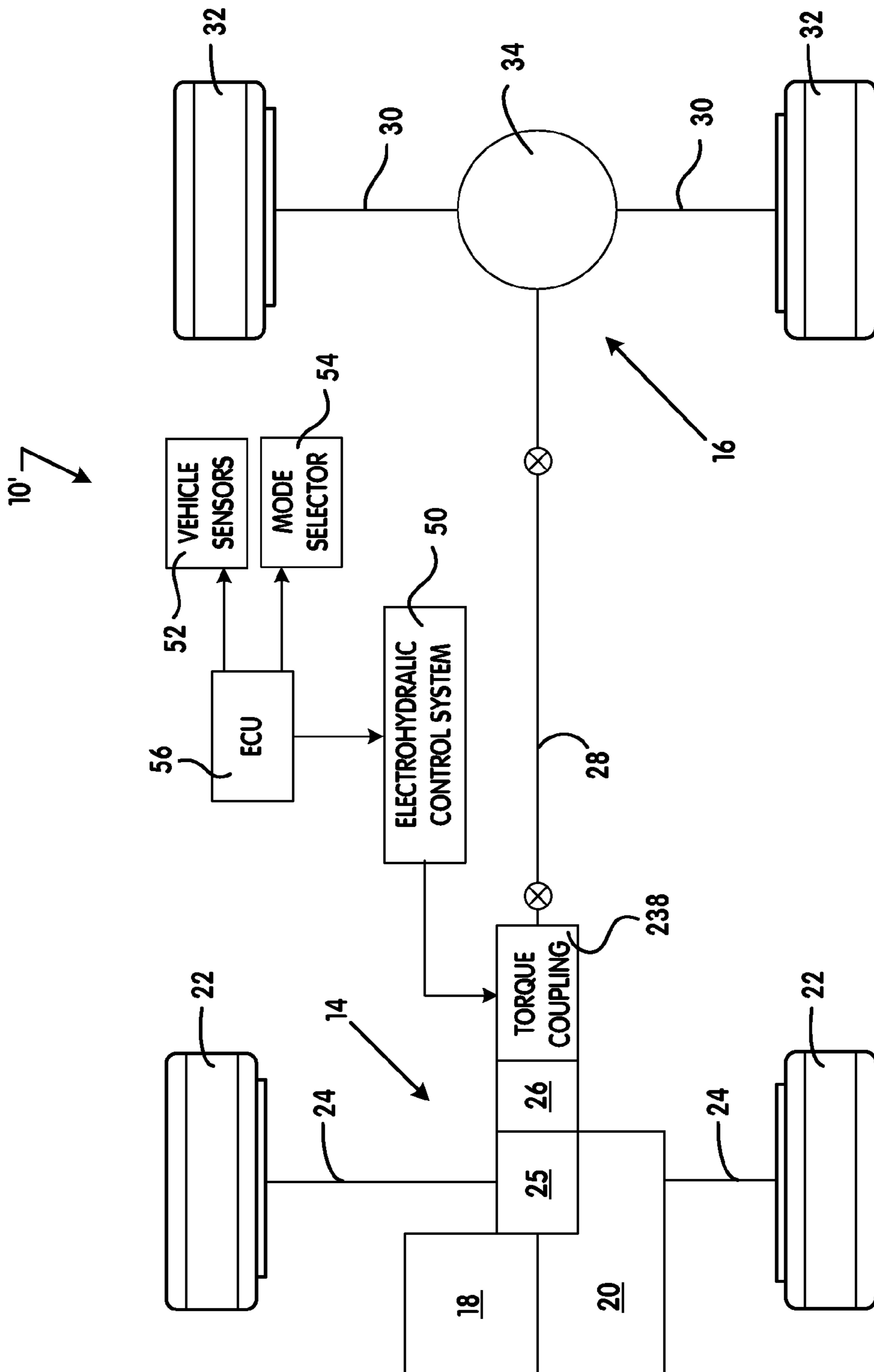


FIG. 11

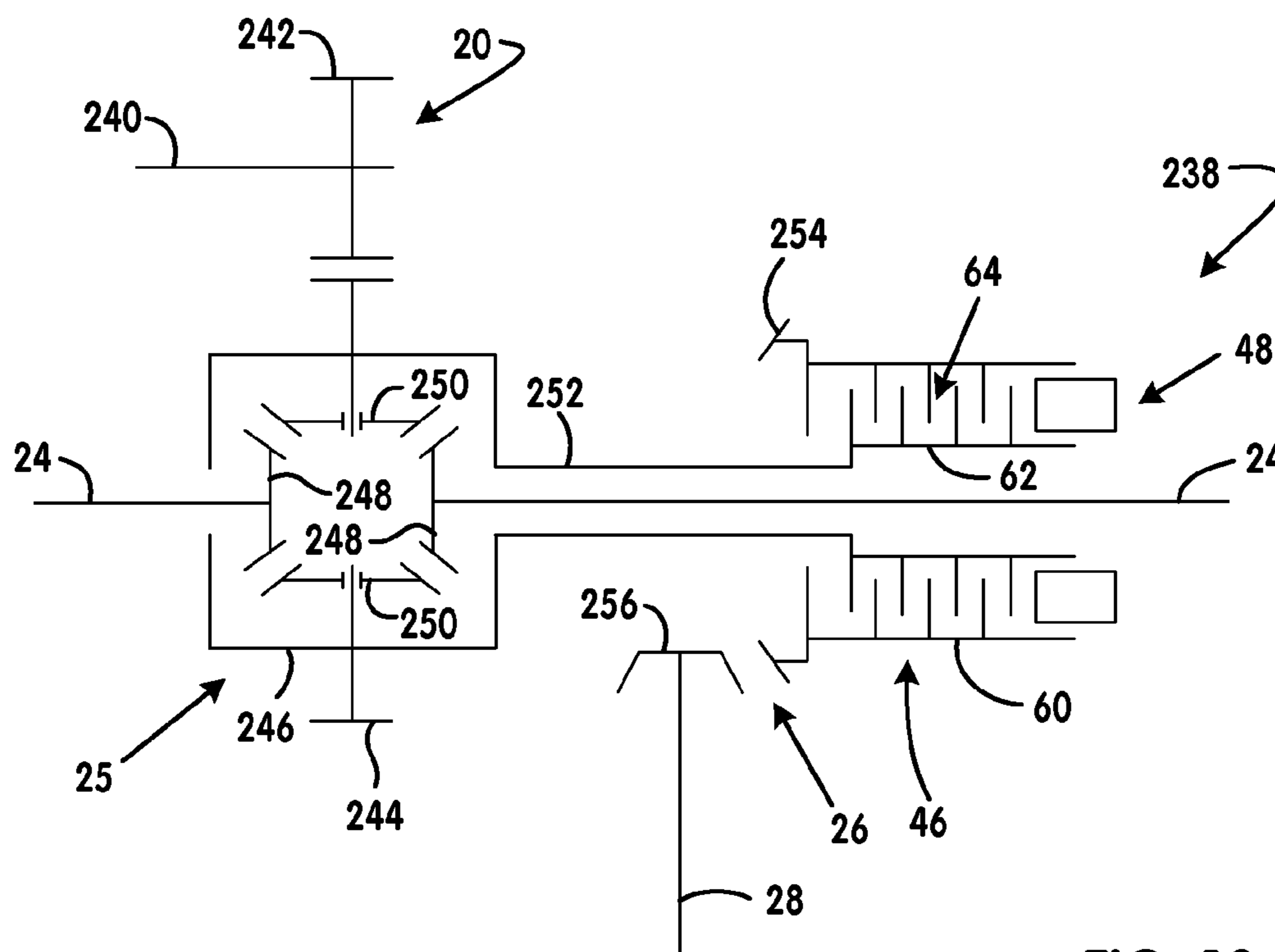


FIG. 12

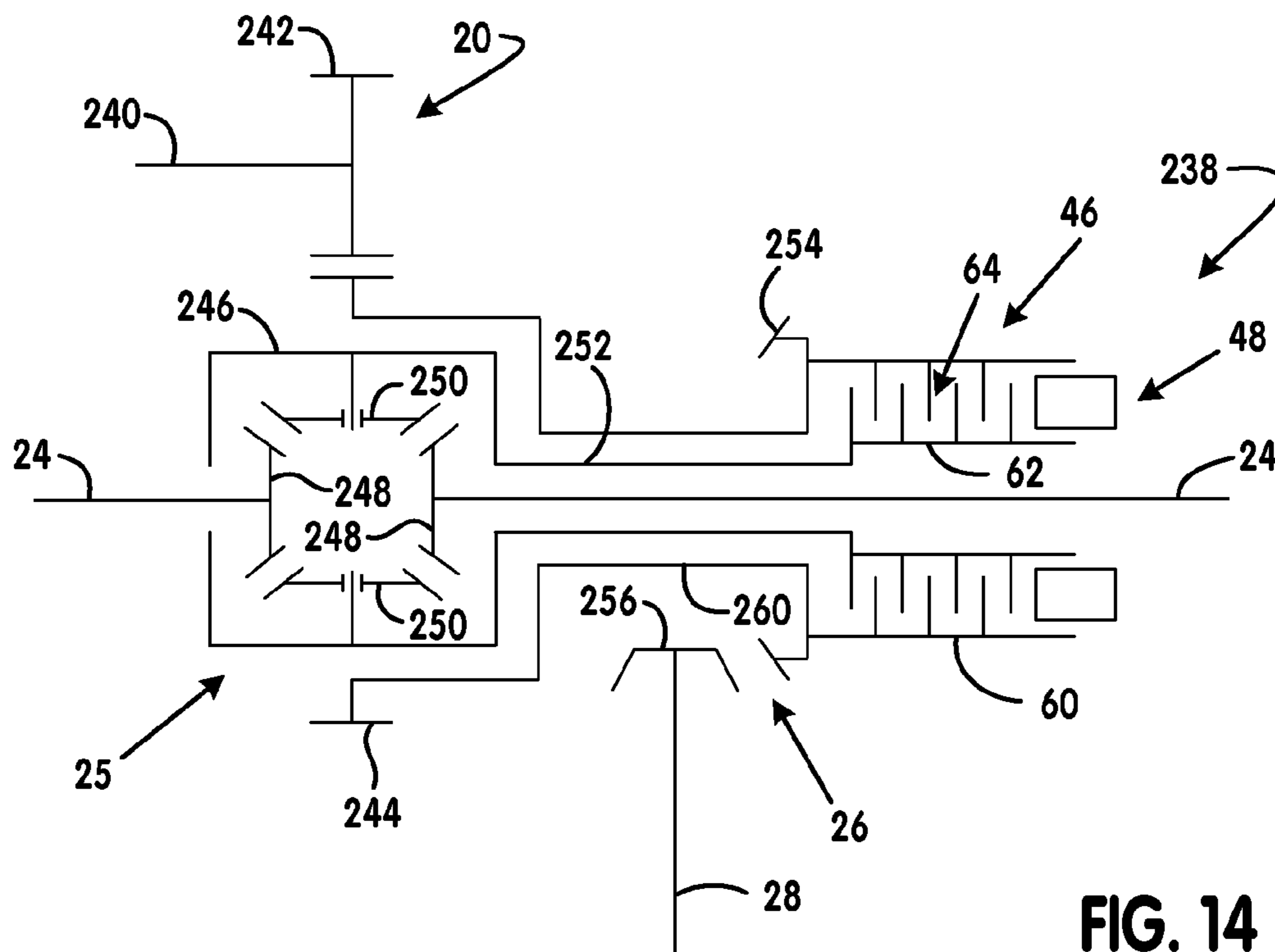


FIG. 14

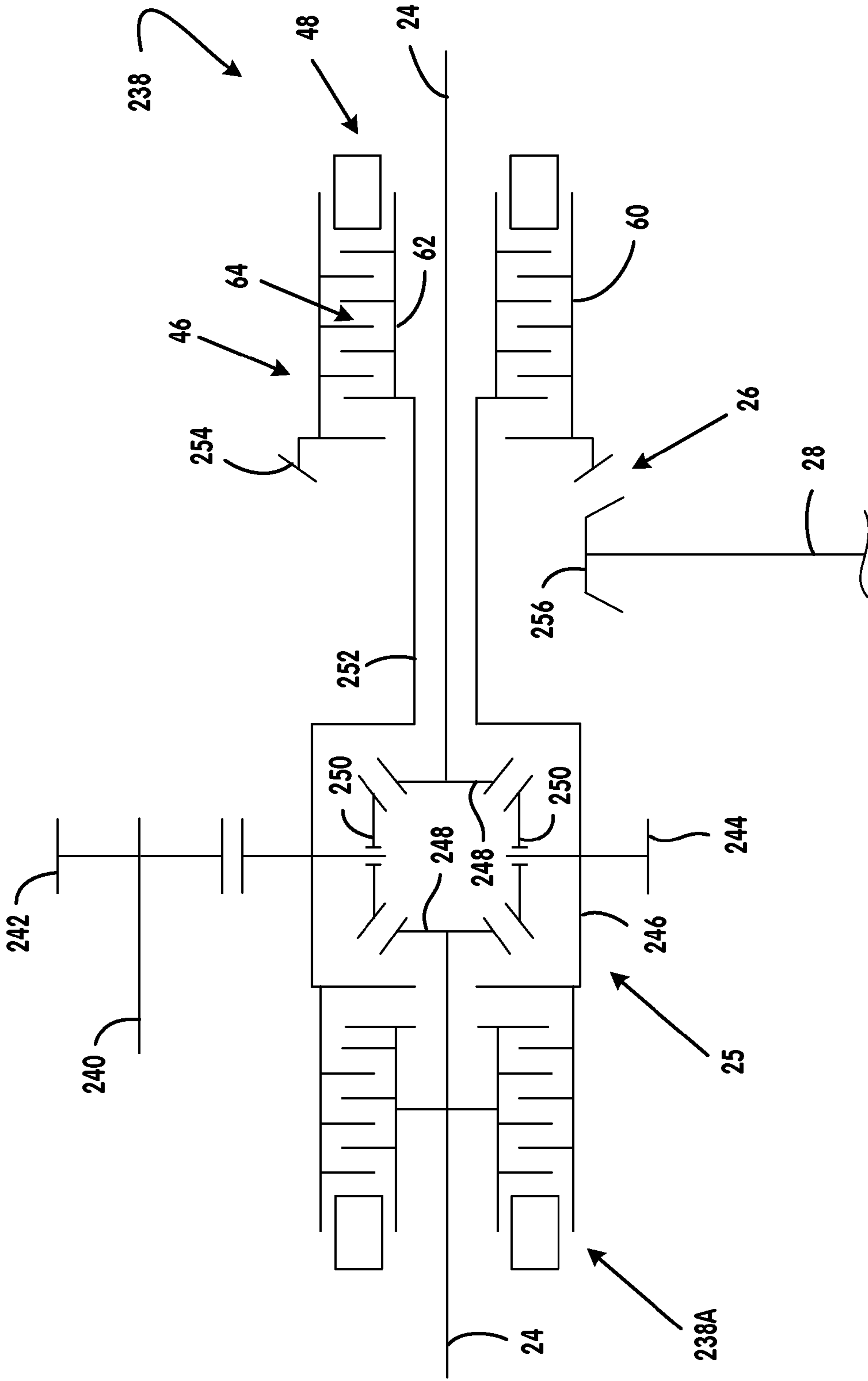


FIG. 13

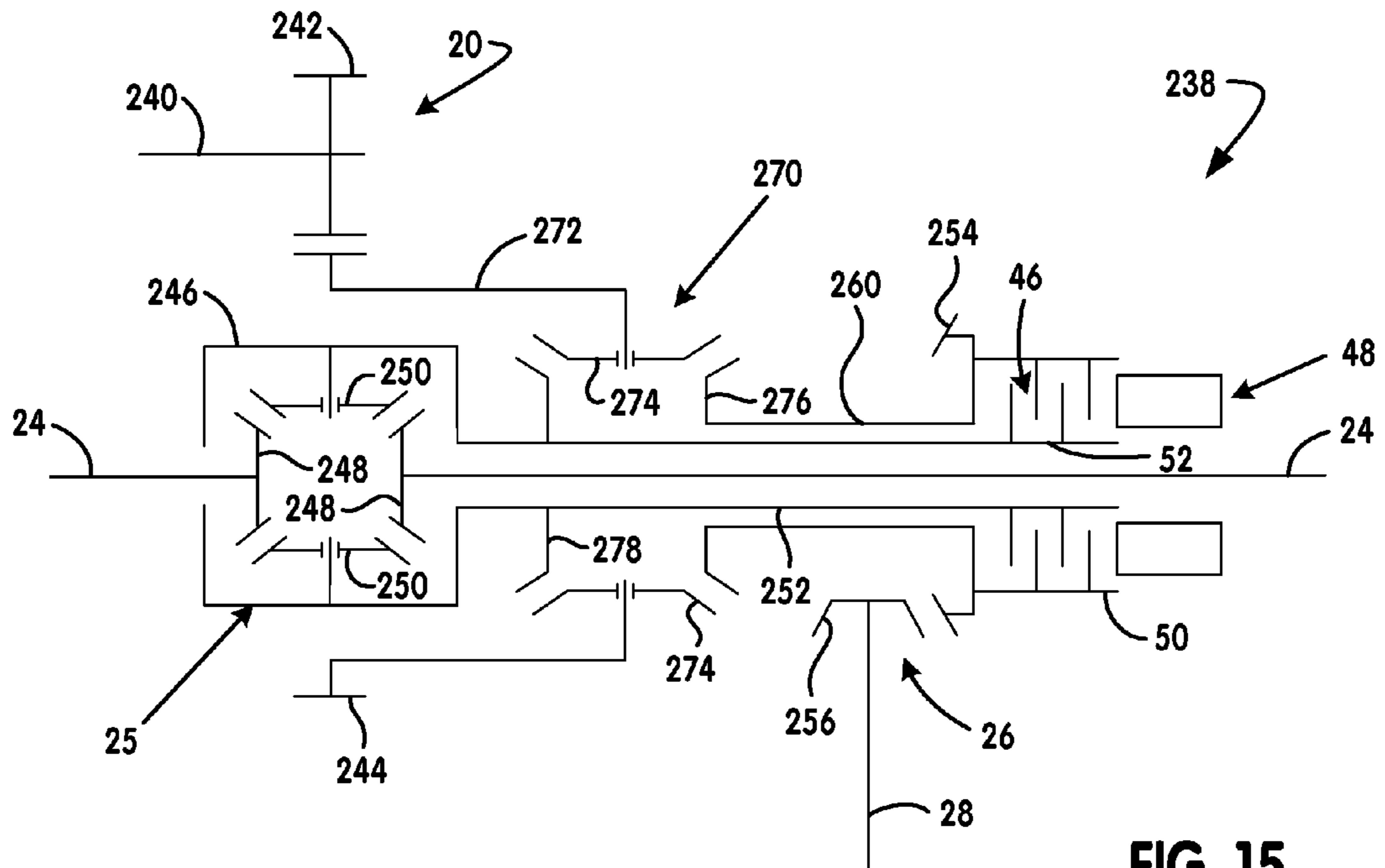


FIG. 15

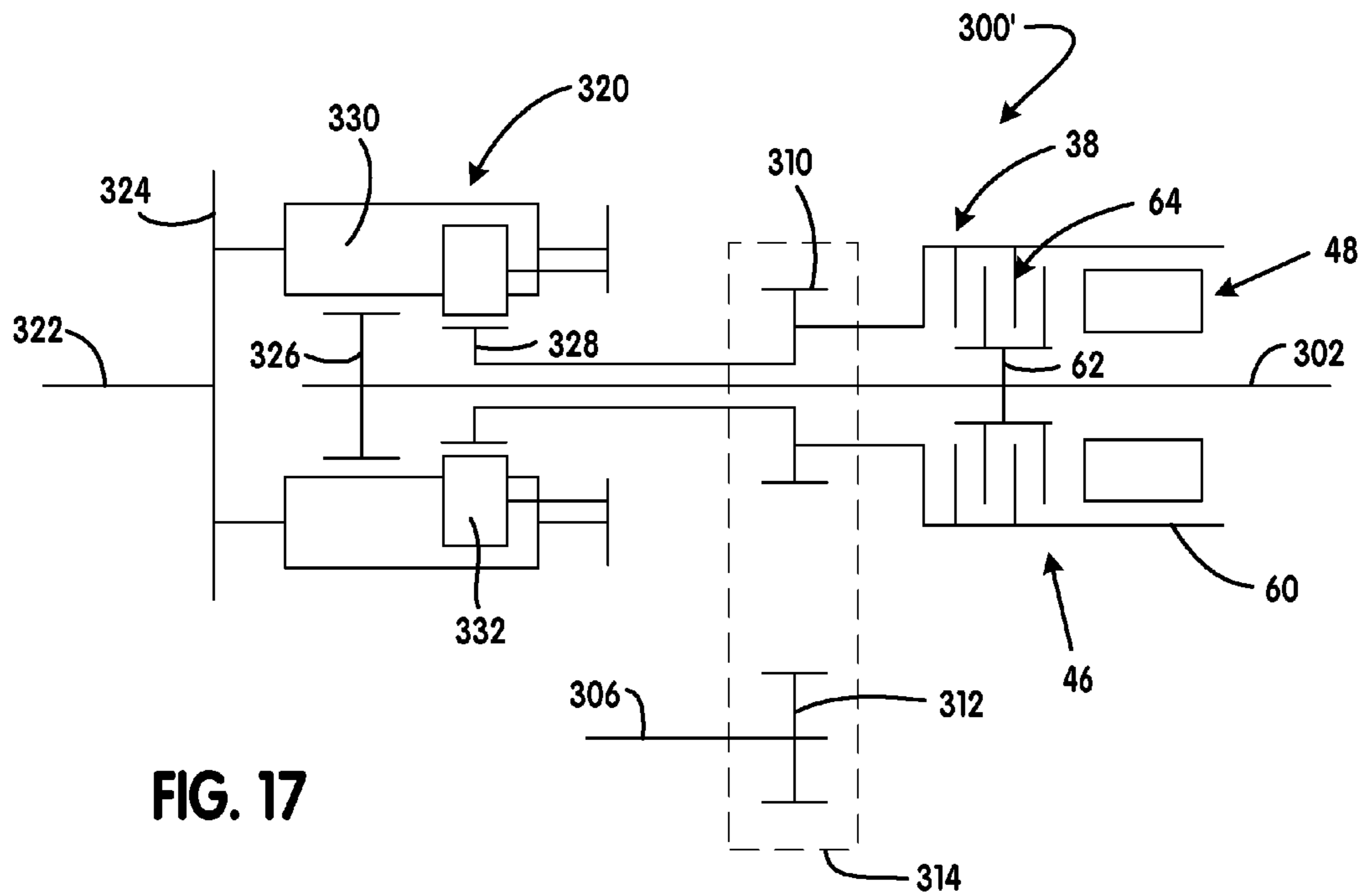


FIG. 17

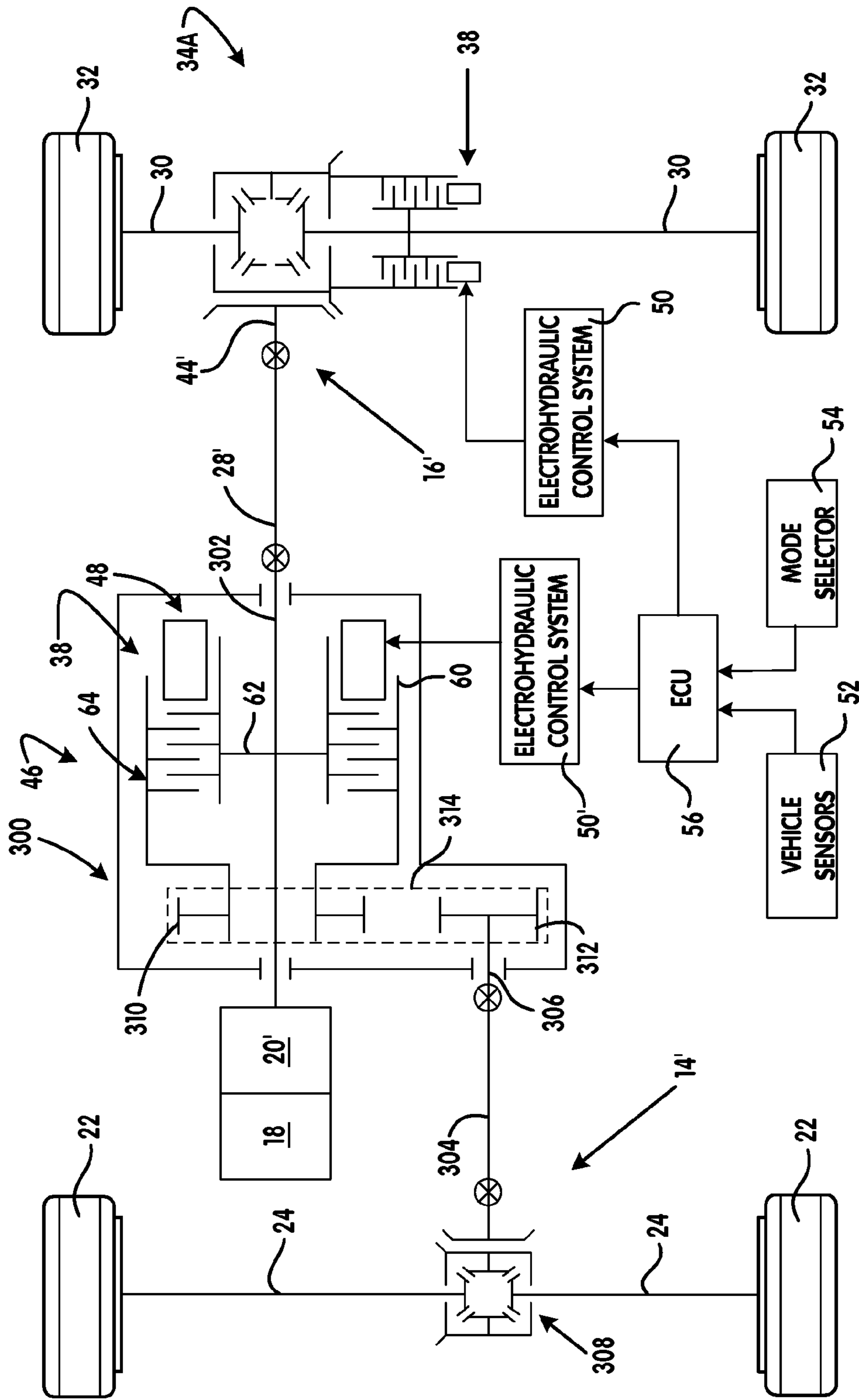


FIG. 16

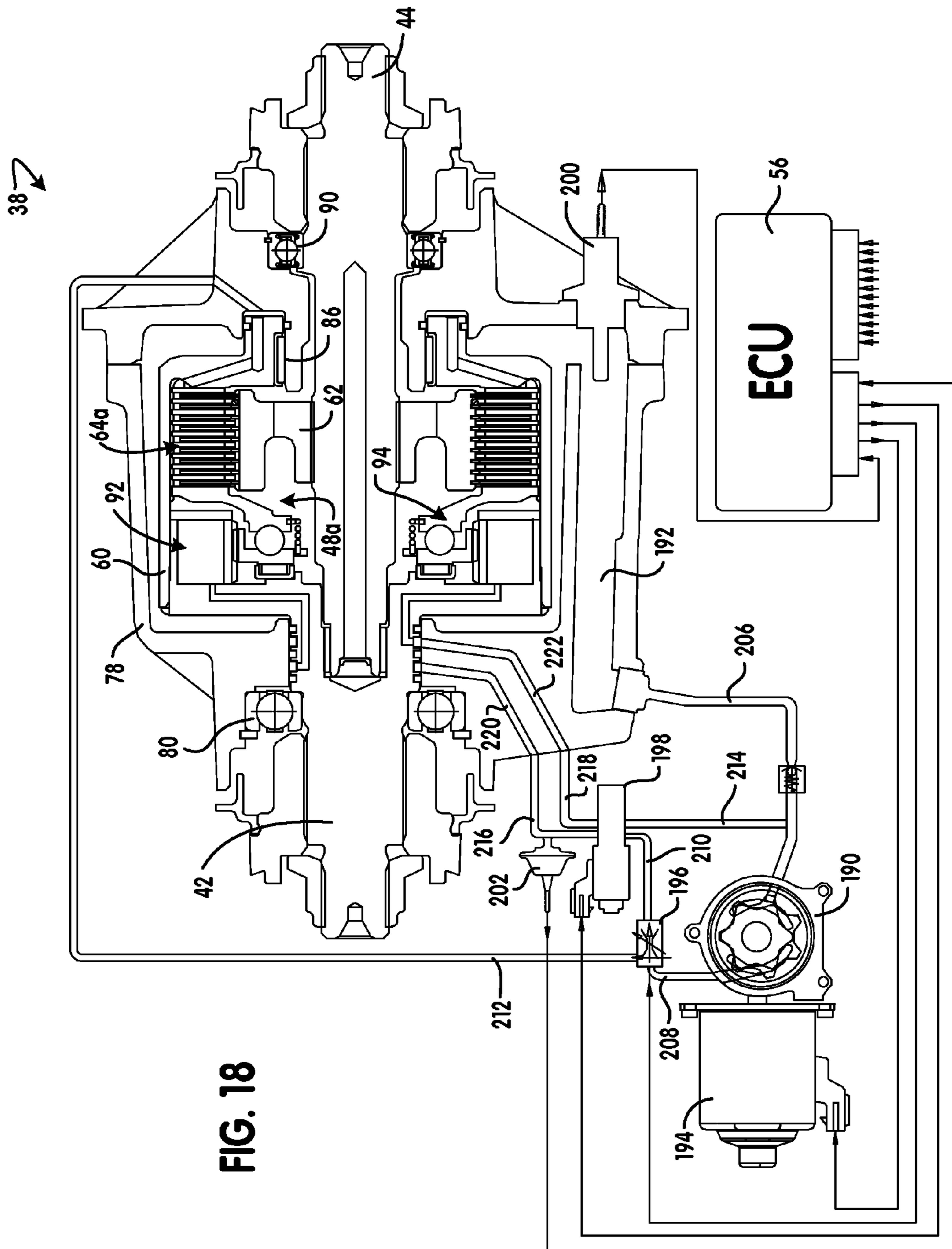


FIG. 18

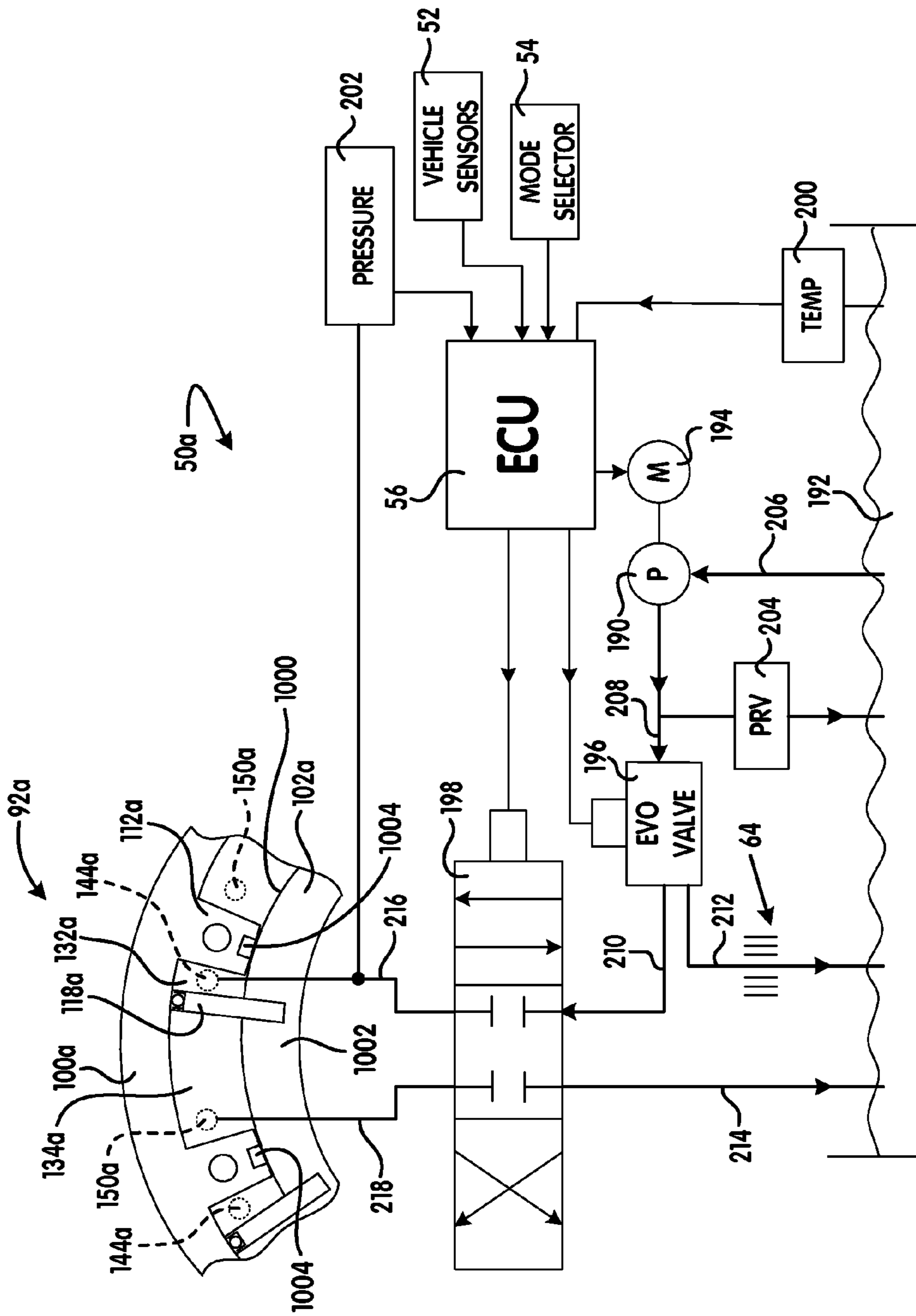


FIG. 19

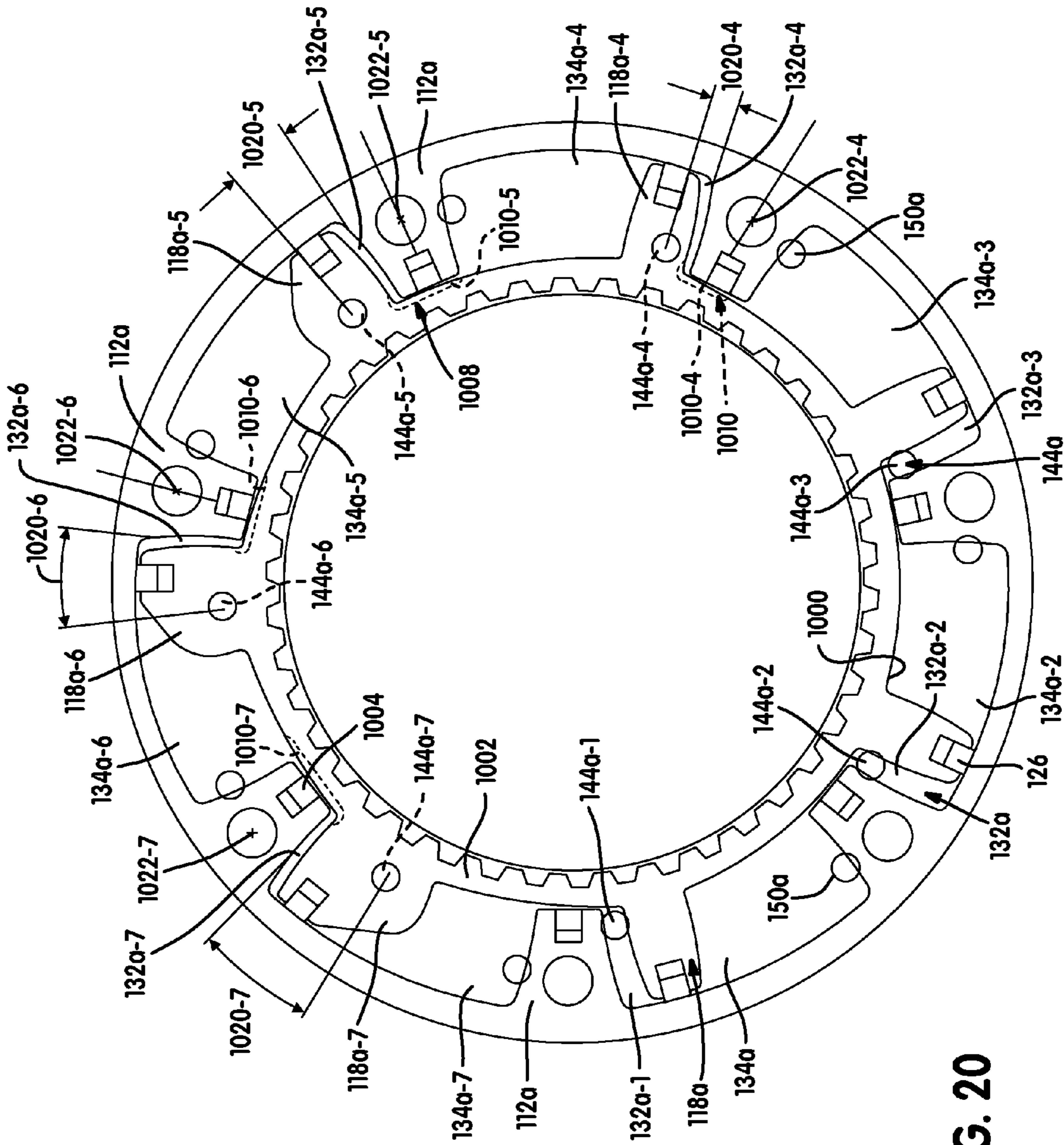


FIG. 20

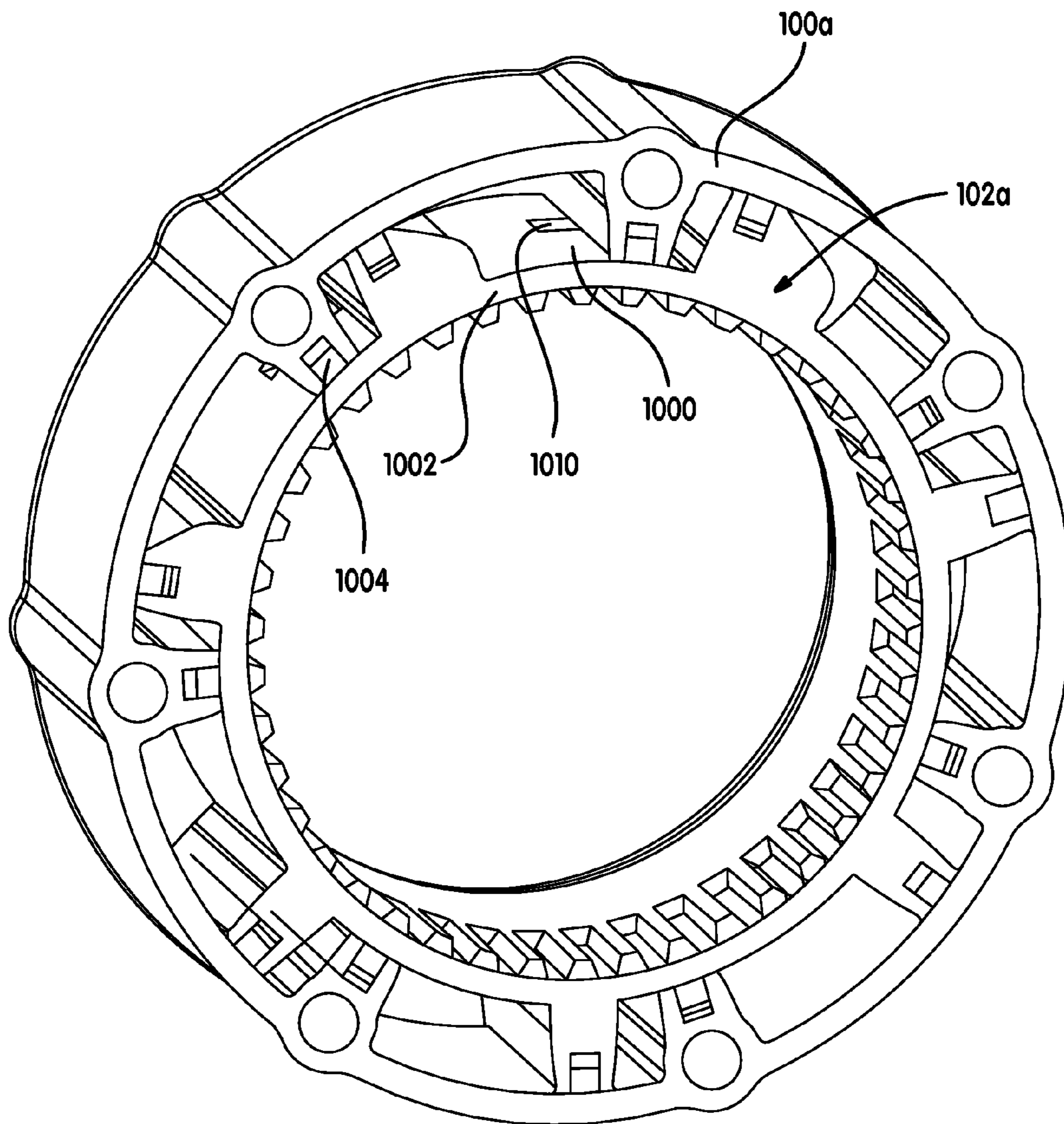


FIG. 21

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**ACTIVE TORQUE COUPLING WITH
HYDRAULICALLY-ACTUATED ROTARY
CLUTCH ACTUATOR**

CROSS-REFERENCE TO RELATED
APPLICATIONS

This application is a continuation-in-part of U.S. patent application Ser. No. 10/771,664 filed on Feb. 4, 2004, now U.S. Pat. No. 6,945,374.

FIELD OF THE INVENTION

The present invention relates generally to power transfer systems for controlling the distribution of drive torque between the front and rear drivelines of a four-wheel drive vehicle and/or the left and right wheels of an axle assembly. More particularly, the present invention is directed to a power transmission device for use in motor vehicle driveline applications having a torque transfer mechanism equipped with a clutch actuator that is operable for controlling actuation of a multi-plate friction clutch.

BACKGROUND OF THE INVENTION

In view of increased demand for four-wheel drive vehicles, a plethora of power transfer systems are currently being incorporated into vehicular driveline applications for transferring drive torque to the wheels. In many vehicles, a power transmission device is operably installed between the primary and secondary drivelines. Such power transmission devices are typically equipped with a torque transfer mechanism for selectively and/or automatically transferring drive torque from the primary driveline to the secondary driveline to establish a four-wheel drive mode of operation. For example, the torque transfer mechanism can include a dog-type lock-up clutch that can be selectively engaged for rigidly coupling the secondary driveline to the primary driveline to establish a "part-time" four-wheel drive mode. When the lock-up clutch is released, drive torque is only delivered to the primary driveline for establishing a two-wheel drive mode.

A modern trend in four-wheel drive motor vehicles is to equip the power transmission device with an adaptively controlled transfer clutch in place of the lock-up clutch. The transfer clutch is operable for automatically directing drive torque to the secondary wheels, without any input or action on the part of the vehicle operator, when traction is lost at the primary wheels for establishing an "on-demand" four-wheel drive mode. Typically, the transfer clutch includes a multi-plate clutch assembly that is installed between the primary and secondary drivelines and a clutch actuator for generating a clutch engagement force that is applied to the clutch assembly. The clutch actuator can be a power-operated device that is actuated in response to electric control signals sent from an electronic controller unit (ECU). The electric control signals are typically based on changes in current operating characteristics of the vehicle (i.e., vehicle speed, interaxle speed difference, acceleration, steering angle, etc.) as detected by various sensors. Thus, such "on-demand" transfer clutch can utilize adaptive control schemes for automatically controlling torque distribution during all types of driving and road conditions. Such adaptively controlled transfer clutches can also be used in association with a center differential operably installed between the primary and secondary drivelines for automatically controlling interaxle slip and torque biasing in a full-time four-wheel drive application.

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A large number of adaptively controlled transfer clutches have been developed with an electro-mechanical clutch actuator that can regulate the amount of drive torque transferred to the secondary driveline as a function of the electric control signal applied thereto. In some applications, the transfer clutch employs an electromagnetic clutch as the power-operated clutch actuator. For example, U.S. Pat. No. 5,407,024 discloses a electromagnetic coil that is incrementally activated to control movement of a ball-ramp drive assembly for applying a clutch engagement force to the multi-plate clutch assembly. Likewise, Japanese Laid-open Patent Application No. 62-18117 discloses a transfer clutch equipped with an electromagnetic clutch actuator for directly controlling actuation of the multi-plate clutch pack assembly. As an alternative, the transfer clutch can employ an electric motor and a mechanical drive assembly as the power-operated clutch actuator. For example, U.S. Pat. No. 5,323,871 discloses a transfer clutch equipped with an electric motor that controls rotation of a sector plate which, in turn, controls pivotal movement of a lever arm that is operable for applying the clutch engagement force to the multi-plate clutch assembly. Likewise, Japanese Laid-open Patent Application No. 63-66927 discloses a transfer clutch which uses an electric motor to rotate one cam plate of a ball-ramp operator for engaging the multi-plate clutch assembly. Finally, U.S. Pat. Nos. 4,895,236 and 5,423,235 respectively disclose a transfer clutch having an electric motor which drives a reduction gearset for controlling movement of a ball screw operator and a ball-ramp operator which, in turn, apply the clutch engagement force to the clutch assembly.

In contrast to the electro-mechanical clutch actuators discussed previously, it is also well known to equip the transfer clutch with an electro-hydraulic clutch actuator. For example, U.S. Pat. Nos. 4,862,769 and 5,224,906 generally disclose use of an electric motor or solenoid to control the fluid pressure exerted by an apply piston on a multi-plate clutch assembly. In addition, U.S. Pat. No. 6,520,880 discloses a hydraulic actuation system for controlling the fluid pressure supplied to a hydraulic motor arranged which is associated with a differential gear mechanism in a drive axle assembly.

While many adaptive clutch actuation systems similar to those described above are currently used in four-wheel drive vehicles, a need exists to advance the technology and address recognized system limitations. For example, the size and weight of the friction clutch components and the electrical power requirements of the clutch actuator needed to provide the large clutch engagement loads make many systems cost prohibitive for use in most four-wheel drive vehicle applications. In an effort to address these concerns, new technologies are being developed for use in power-operated clutch actuator applications.

SUMMARY OF THE INVENTION

Thus, its is an objective of the present invention to provide a power transmission device for use in a motor vehicle having a torque transfer mechanism equipped with a unique electrohydraulically-operated clutch actuator that is operable to control adaptive engagement of a multi-plate clutch assembly.

As a related objective of the present invention, the torque transfer mechanism is well-suited for use in motor vehicle driveline applications to adaptively control the transfer of drive torque between first and second rotary members.

According to each preferred embodiment of the present invention, a torque transfer mechanism and an electrohydraulic control system are disclosed for adaptively controlling the transfer of drive torque between first and second rotary members in a power transmission device of the type used in motor vehicle driveline applications. The torque transfer mechanism includes a rotary input member that is configured to receive drive torque from a source of drive torque, a rotary output member adapted to transmit drive torque to an output device and a torque transmission mechanism that is configured to transfer drive torque from said input member to said output member. The torque transmission mechanism can include a friction clutch that can be operably disposed between the input member and the output member and an electrohydraulic control system for controlling engagement of the friction clutch. The electrohydraulic control system can include a clutch actuator, a fluid pump, a motor driving the fluid pump, and a controller. The clutch actuator can include a rotary operator and a thrust mechanism. The rotary operator can have first and second components with a plurality of actuation chambers and a plurality of return chambers. The first component can be fixed for rotation with one of the input and output members and the second component can be configured to rotate relative to the first component. The thrust mechanism can be operable for applying a clutch engagement force on the friction clutch in response to rotation of the second component relative to the first component. The controller can be operable for controlling the fluid pressure supplied to at least one of the actuation chambers and the return chambers to thereby control rotation of the second component relative to the first component of the rotary operator. The electrohydraulic control system is configured so that the actuation chambers are divided into at least two groups. A first group is configured such that the activation chamber or activation chambers included therein contain a relatively high pressure fluid when the second component is rotated relative to the first component in an actuating direction regardless of a relative position of the first and second components. A second group is configured such that the activation chamber or activation chambers included therein contain a relatively low pressure fluid when the second component is positioned relative to the first component within a predetermined actuating interval and contain the relatively high pressure fluid when the second component is rotated relative to the first component in the actuation direction by an amount that exceeds the predetermined actuating interval.

BRIEF DESCRIPTION OF THE DRAWINGS

Further objectives, features and advantages of the present invention will become apparent to those skilled in the art from analysis of the following written description, the appended claims, and the accompanying drawings in which:

FIG. 1 illustrates the drivetrain of a four-wheel drive vehicle equipped with a power transmission device according to the present invention;

FIG. 2 is a schematic illustration of the power transmission device equipped with a torque transfer mechanism embodying the inventive concepts of the present invention;

FIGS. 3 and 3A are sectional views of the torque transfer mechanism constructed in accordance with a preferred embodiment of the present invention;

FIG. 4 is a sectional view of the rotary operator associated with the torque transfer mechanism shown in FIGS. 3 and 3A;

FIG. 5 is a schematic diagram of the electrohydraulic control circuit associated with the torque transfer mechanism of the present invention;

FIGS. 6 through 8 are schematic diagrams of alternative electrohydraulic control systems adapted for use with the torque transfer mechanism of the present invention;

FIG. 9 is a schematic illustration of an alternative power transmission device available for use with the drivetrain shown in FIG. 1;

FIG. 10 is a schematic illustration of another alternative embodiment of a power transmission device according to the present invention;

FIG. 11 illustrates an alternative drivetrain arrangement for a four-wheel drive motor vehicle equipped with another power transmission device embodying the present invention;

FIGS. 12 through 15 schematically illustrate different embodiments of the power transmission device shown in FIG. 11;

FIG. 16 is an illustration of another drivetrain arrangement for a four-wheel drive vehicle equipped with a power transmission device embodying the present invention;

FIG. 17 is a schematic illustration of an alternative construction for the power transmission device shown in FIG. 16;

FIG. 18 is a sectional view of the torque transfer mechanism constructed in accordance with the teachings of the present invention;

FIG. 19 is a schematic illustration of the power transmission device equipped with the torque transfer mechanism of FIG. 18;

FIG. 20 is a perspective view of a portion of the torque transfer mechanism of FIG. 18 illustrating the clutch actuator in greater detail; and

FIG. 21 is a sectional view of a portion of the torque transfer mechanism of FIG. 18 illustrating the construction of the reaction ring and the actuator ring.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The present invention is directed to a torque transfer mechanism that can be adaptively controlled for modulating the torque transferred from a first rotary member to a second rotary member. The torque transfer mechanism finds particular application in power transmission devices for use in motor vehicle drivelines such as, for example, a torque transfer clutch in a transfer case, a power take-off unit or an in-line torque coupling, a torque biasing clutch associated with a differential unit in full-time transfer cases or power take-off unit or in a drive axle assembly, or any other possible torque transfer application. Thus, while the present invention is hereinafter described in association with particular power transmission devices for use in specific driveline applications, it will be understood that the arrangements shown and described are merely intended to illustrate embodiments of the present invention.

With reference to FIG. 1, a schematic layout of a vehicular drivetrain 10 is shown to include a powertrain 12, a first or primary driveline 14 driven by powertrain 12, and a second or secondary driveline 16. Powertrain 12 includes an engine 18 and a multi-speed transaxle 20 arranged to normally provide motive power (i.e., drive torque) to a pair of first wheels 22 associated with primary driveline 14. Primary driveline 14 further includes a pair of axleshafts 24 connecting wheels 22 to a differential unit 25 associated with transaxle 20. Secondary driveline 16 includes a power

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take-off unit (PTU) 26 driven by the output of transaxle 20, a propshaft 28 driven by PTU 26, a pair of axleshafts 30 connected to a pair of second wheels 32, and a drive axle assembly 34 that is operable to selectively transfer drive torque from propshaft 28 to axle halfshafts 30.

Drive axle assembly 34 is a power transmission device according to one preferred embodiment of the present invention. In particular, drive axle assembly 34 is shown schematically in FIG. 2 to include a housing assembly 36 which encloses a torque transfer mechanism 38 and a differential unit 40. Torque transfer mechanism 38 functions to selectively transfer drive torque from propshaft 28 to an input component of differential unit 40. Specifically, the torque transfer mechanism, hereinafter referred to as torque coupling 38, includes an input shaft 42 driven by propshaft 28, a pinion shaft 44, a transfer clutch 46 operably connected between input shaft 42 and pinion shaft 44, and a clutch actuator 48 for engaging transfer clutch 46.

With continued reference to the drawings, drivetrain 10 is shown to further include an electronically-controlled power transfer system for permitting a vehicle operator to select between a two-wheel drive mode, a locked (“part-time”) four-wheel drive mode, and an adaptive (“on-demand”) four-wheel drive mode. In this regard, transfer clutch 46 can be selectively engaged for transferring drive torque from input shaft 42 to pinion shaft 44 for establishing both of the part-time and on-demand four-wheel drive modes. The power transfer system includes an electrohydraulic control system 50 for selectively actuating clutch actuator 48, vehicle sensors 52 for detecting certain dynamic and operational characteristics of the motor vehicle, a mode select mechanism 54 for permitting the vehicle operator to select one of the available drive modes, and an electronic control unit (ECU) 56 for controlling operation of the components associated with electrohydraulic control system 50 which, in turn, controls actuation of clutch actuator 48 in response to input signals from vehicle sensors 52 and mode selector 54.

Referring primarily to FIGS. 2 and 3, transfer clutch 46 generally includes a first clutch member 60 driven by input shaft 42, a second clutch member 62 driving pinion shaft 44, and a multi-plate clutch pack 64 of alternately interleaved clutch plates installed between input shaft 42 and pinion shaft 44. As shown in this particular arrangement, first clutch member 60 is a clutch drum fixed for rotation with input shaft 42 and second clutch member 62 is a clutch hub fixed (i.e., splined) for rotation with pinion shaft 44. Pinion shaft 44 drives a pinion gear 66 meshed with a ring gear 68 which, in turn, drives differential unit 40. Differential unit 40 includes a carrier 70 driven by ring gear 68, a pair of pinion gears 72 rotatably supported on pinion posts 74 fixed to carrier 70, and a pair of side gears 76. Side gears 76 are meshed with both pinion gears 74 and are coupled for rotation with a corresponding one of axleshafts 30.

Referring now to FIGS. 3 and 3A, a preferred construction for torque coupling 38 will now be described in greater detail. Torque coupling 38 includes a case assembly 78 that is mounted in or forms part of housing 36 of drive axle assembly 34. A bearing assembly 80 supports input shaft 42 for rotation relative to case assembly 78. In addition, input shaft 42 is shown to include an integral end plate 82 that is rigidly secured (i.e., welded) to clutch drum 60. An end plate segment 84 of drum 60 is rotatively supported by a bearing assembly 86 from case assembly 78. Pinion shaft 44 has a first end rotatably supported by a bushing or bearing assembly 88 in a central bore formed in input shaft 42 while its second end extends out of end plate segment 84 of drum 60 and is rotatably supported from case assembly 78 by a

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bearing assembly 90. Clutch actuator 48 generally includes a rotary operator 92, a thrust mechanism 94, and an apply plate 96. Apply plate 96 is secured (i.e., splined) for rotation with drum 60 of transfer clutch 46.

As will be detailed, clutch actuator 48 is operable for generating and exerting a compressive clutch engagement force on clutch pack 64. Such engagement of clutch pack 64 causes rotary power (“drive torque”) to be transferred from input shaft 42 to pinion shaft 44. Specifically, clutch actuator 48 is operable for controlling axial movement of apply plate 96 and thus, the magnitude of the clutch engagement force applied to clutch pack 64. In particular, apply plate 96 is axially moveable relative to clutch pack 64 between a first or “released” position and a second or “locked” position. With apply plate 96 in its released position, a minimum clutch engagement force is exerted on clutch pack 64 such that virtually no drive torque is transferred from input shaft 42 through transfer clutch 46 to pinion shaft 44, thereby establishing the two-wheel drive mode. In contrast, movement of apply plate 96 to its locked position causes a maximum clutch engagement force to be applied to clutch pack 64 such that pinion shaft 44 is, in effect, coupled for common rotation with input shaft 42, thereby establishing part-time four-wheel drive mode. Accordingly, controlling the position of apply plate 96 between its released and locked positions permits adaptive regulation of the amount of drive torque transferred from input shaft 42 to pinion shaft 44, thereby establishing the on-demand four-wheel drive mode.

Rotary operator 92 includes a first or reaction ring 100 that is concentrically aligned with a second or “actuator” ring 102. The rings are retained in an annular chamber for 104 defined between end plate 82 and a retainer plate 106. While not shown, retainer plate 106 is secured by a plurality of bolts to end plate 82 which also extend through mounting holes 108 in reaction ring 100. As such, reaction ring 100 is fixed for common rotation with input shaft 42.

As best seen from FIG. 4, reaction ring 100 includes a cylindrical body segment 110 and a plurality of radially inwardly projecting lugs 112 which define a complimentary number of longitudinally extending channels 114 therebetween. Likewise, actuator ring 102 has a cylindrical body segment 116 and a plurality of separator plates 118 that are retained in longitudinal slots 120 formed in body segment 116. An outer edge surface 122 of each separator plate 118 is aligned to be in sliding engagement with an inner wall surface 124 of channels 114. A seal strip 126 may be attached to edge surface 122 to provide a sliding fluid-tight interface between separator plates 118 and body segment 110 of reaction ring 100. Likewise, a terminal edge surface 128 of each lug 112 is aligned to be in sliding engagement with an outer wall surface 130 of actuator ring 102. Each separator plate 118 defines an actuation chamber 132 and a return chamber 134 between adjacent pairs of lugs 112. As such, a plurality of circumferentially-spaced alternating actuation chambers 132 and return chambers 134 are established in association with rotary operator 92.

To provide means for supplying hydraulic fluid from electrohydraulic control system 50 to actuation chambers 132, a first flow path is formed in input shaft 42 and its end plate segment 82. The first flow passage includes an annular chamber 140 which communicates with a series of circumferentially-spaced flow passages 142 having ports 144 in fluid communication with actuation chambers 132. Similarly, a second flow path provides means for supplying hydraulic fluid from control system 50 to return chambers 134. This second flow path includes an annular chamber 146

which communicates with a series of circumferentially-spaced flow passages **148** having ports **150** in fluid communication with return chambers **134**. As will be detailed, increasing the fluid pressure delivered through ports **144** to actuation chambers **132** while decreasing the fluid pressure delivered through ports **150** to return chambers **134** causes actuator ring **102** to move (i.e., index) in a first rotary direction (i.e., counterclockwise) relative to reaction ring **100** for causing thrust mechanism **94** to urge apply plate **96** to move toward its locked position. In contrast, decreasing the fluid pressure in actuation chambers **132** and increasing the fluid pressure in return chambers **134** causes actuator ring **102** to index in a second rotary direction (i.e., clockwise) relative to reaction ring **100** for causing thrust mechanism **94** to permit apply plate **96** to move toward its released position.

With continued reference primarily to FIG. 3A, thrust mechanism **94** is shown to be a ball ramp unit having a first cam member **152**, a second cam member **154**, and rollers, such as balls **156**. First cam member **152** is fixed via a spline connection **158** for rotary movement with actuator ring **102**. Likewise, second cam member **154** is fixed to apply plate **96** for common rotation with drum **60** and input shaft **42**. Each ball **156** is disposed in a cam channel defined between a cam track **170** formed in first cam member **152** and a corresponding cam track **172** formed in second cam member **154**. Preferably, a plurality of cam channels are provided between first cam member **152** and second cam member **154** with cam tracks **170** and **172** configured as oblique sections of a helical torus. However, balls **156** and cam tracks **170** and **172** may be replaced with alternative cam components and/or ramp configurations that function to cause axial displacement of second cam member **154**. In any arrangement, the load transferring components can not be self-locking or self-engaging so as to permit fine control of the translational movement of apply plate **96** (via second cam member **154**) for precise control of the engagement characteristics (i.e., torque transfer) of transfer clutch **46**. As seen, a thrust bearing **176** is located between end plate **82** and first cam member **152**.

Ball ramp unit **94** further includes a torsional return spring **178** that is operably disposed between cam members **152** and **154**. Return spring **178** functions to angularly bias cam members **152** and **154** to return to a "retracted" position for de-energizing ball ramp unit **94**. Such angular movement of the cam members to the retracted position due to the biasing of return spring **178** results in angular movement of actuator ring **102** relative to reaction ring **100** in the second angular direction toward a first or "low pressure" position and translational movement of apply plate **96** toward its released position. With actuator ring **102** in its low pressure position (as shown in FIG. 4), ball ramp unit **94** is de-energized and apply plate **96** is in its released position so as to exert a predetermined minimum clutch engagement force on clutch pack **64** for releasing engagement of transfer clutch **46**.

Electrohydraulic control system **50** is operable to supply high pressure fluid to actuation chambers **132** for causing actuator ring **102** to rotate relative to reaction ring **100** in the first direction from its low pressure position toward a second or "high pressure" position. Such movement of actuator ring **102** results in corresponding relative angular movement between cam members **152** and **154** from the retracted position toward a second or "extended" position for energizing ball ramp unit **94**. Accordingly, the profile of cam tracks **170** and **172** establishes the resultant amount of translational movement of second cam member **154** required to cause corresponding axial movement of apply plate **96**

from its released position toward its locked position. When actuator ring **102** is in its high pressure position, ball ramp unit **94** is fully energized and apply plate **96** is located in its locked position such that the maximum clutch engagement force is exerted on clutch pack **64** for fully engaging transfer clutch **46**. Furthermore, electrohydraulic control system **50** is operable to supply high pressure fluid to return chambers **134** and vent actuation chambers **132** for causing actuator ring **102** to rotate relative to reaction ring **100** in the second direction from its high pressure position. Such angular movement of actuator ring **102** results in corresponding relative angular movement between cam members **152** and **154** from the extended position toward the retracted position for releasing ball ramp unit **94**. As such, apply plate **96** is caused to move from its locked position toward its released position for releasing engagement of transfer clutch **46**.

With apply plate **96** in its released position, virtually no drive torque is transferred from input shaft **42** to pinion shaft **44** through torque coupling **38** so as to effectively establish the two-wheel drive mode. In contrast, location of apply plate **96** in its locked position results in a maximum amount of drive torque being transferred to pinion shaft **44** for coupling pinion shaft **44** for common rotation with input shaft **42**, thereby establishing the part-time four-wheel drive mode. Accordingly, controlling the position of apply plate **96** between its released and locked positions permits variable control of the amount of drive torque transferred from input shaft **42** to pinion shaft **44** for establishing an on-demand four-wheel drive mode. Thus, the magnitude of the fluid pressure supplied to actuation chambers **132** and return chambers **134** controls the angular position of actuator ring **102** relative to reaction ring **100**, thereby controlling actuation of ball ramp unit **94** and the resulting movement of apply plate **96** between its released and locked positions.

Referring to FIGS. 3 and 5, the components associated with electrohydraulic control system **50** will now be described. The basic components of control system **50** include a fluid pump **190** operable to draw fluid from a fluid source or sump **192** within casing **78**, an electric motor **194** driving pump **190**, an electronic variable orifice (EVO) valve **196**, and a directional control valve **198**. Control system **50** further includes a temperature sensor **200**, a pressure sensor **202** and a spring-type pressure relief valve (PRV) **204**. Motor **194** is a low power unit operable for causing pump **190** to draw fluid from sump **192** through a first flow path **206**. The magnitude of the line pressure discharged from pump **190** is delivered to an inlet of EVO valve **196** through a second flow path **208**. However, PRV **204** functions to limit the line pressure delivered to the inlet of EVO valve **196** to a predefined maximum fluid pressure value. EVO valve **196** includes a moveable valve element that functions to vary the line pressure to a "control" pressure value that is supplied from an outlet of EVO valve **196** to an inlet of control valve **198** via a third flow path **210**. As seen, the low pressure fluid by-passed through EVO valve **196** flows through a fourth flow path **212** and is used to cool and lubricate clutch pack **64**. Fluid discharged from control valve **198** is returned via a fifth flow path **214** to sump **192**. Finally, control valve **198** is shown to be in fluid communication with actuation chambers **132** via a sixth flow path **216** and in fluid communication with return chambers **134** via a seventh flow path **218**. It will be appreciated that the components of the electrohydraulic control system can be integrated into case assembly **78** to define a stand-alone assembly or, in the alternative, be remotely located and connected via suitable hoses and tubing.

Rotary operator **92** is partially shown in FIG. **5** to illustrate its hydraulic connections with control valve **198**. In the arrangement shown, EVO valve **196** is preferably an electromagnetic flow control valve that is capable of variably regulating the fluid pressure delivered to control valve **198**. Likewise, control valve **198** is shown as a 3-position shuttle valve. As seen, ECU **56** is operable to send electrical control signals to electric motor **194**, EVO valve **196** and control valve **198**. EVO valve **196** is selectively actuated by ECU **56** to control the fluid pressure supplied to third flow path **210** while control valve **198** is selectively actuated to control delivery of the pressurized fluid to either of actuation chambers **132** or return chambers **134**. As best seen from FIGS. **3** and **3A**, sixth flow path **216** is in fluid communication with annular chamber **140** via a flow passage **220** formed in case **78** while seventh flow path **218** communicates with annular chamber **146** via another flow passage **222**. A plurality of ring seals **224** are provided between case **78** and input shaft **42** to delimit chambers **140** and **146**.

As contemplated by the present invention, ECU **56** is programmed to accurately control the angular position of actuator ring **102** relative to reaction ring **100** based on a predefined control strategy for transferring the desired amount of drive torque across transfer clutch **46**. The control strategy functions to determine and generate the electric control signals sent to EVO valve **196** and control valve **198** based on the mode signal from mode selector **54** and the sensor signals from sensors **52**. In addition, pressure sensor **202** sends a signal to ECU **56** that is indicative of the fluid pressure delivered through sixth flow path **216** to actuation chambers **132**. Likewise, temperature sensor **200** sends a signal to ECU **56** that is indicative of the fluid temperature in sump **192**.

In operation, if the vehicle operator selects the two-wheel drive mode, EVO valve **196** is opened and control valve **198** is initially actuated to cause third flow path **210** to be placed in communication with seventh flow path **218** while sixth flow path **216** is placed in communication with fifth flow path **214**, whereby actuation chambers **132** are vented to sump **192** while fluid at control pressure is supplied to return chambers **134**. This results in actuator ring **102** being forced to its low pressure position relative to reaction ring **100** for releasing engagement of transfer clutch **46**. Thereafter, control valve **198** can be shifted into its closed position with no fluid delivered to actuation chambers **132** or return chambers **134** since return spring **178** forcibly biases actuator ring **102** to remain in its low pressure position. In contrast, upon selection of the part-time four-wheel drive mode, EVO valve **196** is opened and control valve **198** is actuated to connect third flow path **210** to sixth flow path **216** and also connect seventh flow path **218** to fifth flow path **214**, whereby actuation chambers **132** are supplied with fluid at control pressure and return chambers **134** are vented to sump **192**. The high pressure fluid supplied to actuation chambers **132** causes actuator ring **102** to move to its high pressure position relative to reaction ring **100**, whereby ball ramp unit **94** is fully energized for moving apply plate **96** to its locked position for fully engaging transfer clutch **46**. PRV **204** functions to limit the maximum fluid pressure that can be delivered to actuation chambers **132** during part-time four-wheel drive operation, thereby providing a torque limiting feature to prevent damage to clutch pack **64**.

When mode selector **54** indicates selection of the on-demand four-wheel drive mode, ECU **56** actuates control valve **198** so as to connect third flow path **210** to sixth flow path **216** while also connecting seventh flow path **218** to fifth flow path **214**. As such, return chambers **134** are vented and

supply actuation chambers **132** are supplied with pressurized fluid from EVO valve **196**. ECU **56** functions to adaptively control EVO valve **196** so as to initially supply fluid at a predetermined relatively low pressure to actuation chambers **132** that causes actuator ring **102** to index slightly relative to reaction ring **100** in the first direction. This angular movement causes actuator ring **102** to move from its low pressure position to an intermediate or "ready" position which, in turn, results in ball ramp unit **94** moving apply plate **96** from its released position to a "stand-by" position. In the stand-by position, apply plate **96** exerts a small clutch engagement force on clutch pack **64**. Accordingly, a small amount of drive torque is delivered to pinion shaft **44** through transfer clutch **46** in this adapt-ready condition. Thereafter, ECU **56** determines when and how much drive torque needs to be transferred to pinion shaft **44** based on the current tractive conditions and/or operating characteristics of the motor vehicle, as detected by sensors **52**.

Sensors **52** detect such parameters as, for example, the rotary speed of the input and pinion shafts, the vehicle speed and/or acceleration, the transmission gear, the on/off status of the brakes, the steering angle, the road conditions, etc. Such sensor signals are used by ECU **56** to determine a desired output torque value utilizing a control logic scheme that is incorporated into ECU **56**. In particular, the control logic correlates the desired torque value to a fluid pressure value to be delivered to actuation chambers **132**. Based on this desired pressure value, ECU **56** actively controls actuation of EVO valve **196** to generate a corresponding pressure level in actuation chambers **132**. Pressure sensor **202** provides ECU **56** with direct feedback as to the actual fluid pressure in actuation chambers **132** so as to permit precise control of clutch actuator **52**.

In addition to adaptive on-demand torque control, the present invention permits automatic release of transfer clutch **46** in the event of an ABS braking condition or during the occurrence of an over-temperature condition. Specifically, when ECU **56** is signaled that an ABS brake condition occurs, control valve **198** is immediately actuated to connect return chambers **134** with EVO valve **196** while actuation chambers **132** are vented to sump **192**. Also, EVO valve **196** is fully opened to send full pressure to return chambers **134**, thereby forcibly moving actuator ring **102** in its second direction to its low pressure position for fully releasing transfer clutch **46**. Moreover, if temperature sensor **200** detects that the fluid temperature in sump **192** exceeds a predetermined threshold value, the same type of immediate release of transfer clutch **46** will be commanded by ECU **56**.

While the control scheme has been described based on an on-demand torque transfer strategy, it is contemplated that a differential "mimic" control strategy can likewise be used. Specifically, the torque distribution between input shaft **42** and pinion shaft **44** can be controlled to normally maintain a predetermined rear/front ratio (i.e., 70:30, 50:50, etc.) so as to simulate the inter-axle torque splitting feature typically provided by a center differential unit. This desired torque distribution can then be adaptively controlled to address lost traction at either set of wheels. Regardless of whether the control logic scheme is based on an on-demand or a differential torque transfer strategy, accurate control of the fluid pressure delivered to actuation chambers **132** of rotary operator **92** provides the desired torque transfer characteristics being established across transfer clutch **46**.

It is contemplated that the 3-position directional control valve **198** shown in FIG. **5** could easily be substituted with a 2-position directional valve or any other known valve device capable of controlling the direction of flow in the

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hydraulic circuit. Likewise, FIG. 6 illustrates a modified arrangement for electrohydraulic control system 50 where control valve 198 now receives the line pressure directly from pump 190 via flow path 208 and EVO valve 196 is functional to regulate the fluid pressure within actuation chambers 132 by controlling the fluid pressure within flow path 216. Again, EVO valve 196 is normally open with the by-passed fluid delivered through flow path 212 to cool and lubricate clutch pack 64. In operation, the fluid pressure supplied to actuation chambers 132 is increased by closing EVO valve 196. As such, variable control of EVO valve 156 again results in variable pressure control within actuation chambers 132.

FIG. 7 is a hydraulic schematic of an electrohydraulic clutch system 50 which is generally similar to FIG. 6 except that control valve 198 has been eliminated. As such, ECU 56 controls the pumping direction of pump 190 by reversing the polarity of motor 194. Check valves 230 and 232 are provided to permit such a bi-directional pumping action. As before, the pressure profile of the fluid delivered to actuation chambers 132 is adaptively controlled via variable actuation of EVO valve 196. Optionally, EVO valve 196 can be eliminated with the pressure profile within actuation chambers 132 and return chambers 134 directly controlled by variable control of the rotary speed (RPM) and direction of electric motor 194.

The previous hydraulic circuits utilized control valve 198 for directional control in conjunction with EVO valve 196 for adaptive pressure regulation. As an alternative, FIG. 8 illustrates a hydraulic circuit wherein a proportional control valve 234 is used to regulate both the directional and pressure characteristics. Preferably, proportional control valve 234 is a pulse width modulated (PWM) valve having a moveable element that is controlled by an electromagnetic solenoid based on electric control signals from ECU 56. In addition, a flow control valve 236 is provided in flow path 208 to prevent dead-heading of pump 190 and to provide cooling flow to clutch pack 64.

The arrangement shown for drive axle assembly 34 of FIG. 2 is operable to provide the on-demand four-wheel drive mode by adaptively transferring drive torque from primary driveline 14 (via propshaft 28) to secondary driveline (via pinion shaft 44). In contrast, a drive axle assembly 34A is shown in FIG. 9 with torque coupling 38 now installed between differential case 70 and one of axleshafts 30 to provide an adaptive system for biasing the torque and limiting intra-axle slip between the rear wheels 32. As before, torque coupling 38 is schematically shown to include transfer clutch 46 and clutch actuator 48, the construction and function of which are understood to be similar to the detailed description previously provided herein for each sub-assembly. It will be understood that this particular "limited slip" differential arrangement can either be used in association with the on-demand drive axle assembly shown in FIG. 2 or in association with a drive axle assembly wherein propshaft 28 directly drives differential unit 40.

Referring now to FIG. 10, a drive axle assembly 34B is schematically shown to include a pair of torque couplings 38L and 38R that are operably installed between a driven shaft 44 or 28 and axleshafts 30. The driven pinion shaft drives a right-angled gearset including pinion 66 and ring gear 68 which, in turn, drives a transfer shaft 240. First torque coupling 38L is shown disposed between transfer shaft 240 and the left axleshafts 30 while second torque coupling 38R is disposed between transfer shaft 240 and the right axle shaft 30. Each coupling includes a corresponding transfer clutch 46L, 46R and a clutch actuator 48L, 48R.

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Accordingly, independent slip control between the driven pinion shaft and each wheel 32 is provided by this arrangement. A common sump is provided with ECU 56 controlling independent actuation of both clutch actuators 48L and 48R.

To illustrate an alternative power transmission device to which the present invention is applicable, FIG. 11 schematically depicts a front-wheel based four-wheel drive drivetrain layout 10' for a motor vehicle. In particular, engine 18 drives a multi-speed transaxle 20 having an integrated front differential unit 25 for driving front wheels 22 via axle shafts 24. As before, PTU 26 is also driven by transaxle 20 for delivering drive torque to the input member of a torque coupling 238. The output member of torque coupling 238 is coupled to propshaft 28 which, in turn, drives rear wheels 32 via axle assembly 34. Rear axle assembly 34 can be a traditional driven axle with a differential or, in the alternative, be similar to the arrangements described in association with FIGS. 9 or 10. Accordingly, in response to the detection of a front wheel slip condition, torque coupling unit 238 is adaptively actuated to deliver drive torque "on-demand" to rear wheels 32. Again, it is contemplated that torque coupling unit 238 is substantially similar in structure and function to that of torque coupling unit 38 previously described herein.

Referring now to FIG. 12, torque coupling 238 is schematically illustrated in association with an on-demand four-wheel drive system based on a front-wheel drive vehicle similar to that shown in FIG. 11. In particular, an output shaft 240 of transaxle 20 is shown to drive an output gear 242 which, in turn, drives an input gear 244 that is fixed to a carrier 246 associated with front differential unit 25. To provide drive torque to front wheels 22, front differential unit 25 includes a pair of side gears 248 that are connected to front wheels 22 via axleshafts 24. Differential unit 25 also includes pinions 250 that are rotatably supported on pinion shafts fixed to carrier 246 and which are meshed with side gears 248. A transfer shaft 252 is provided for transferring drive torque from carrier 246 to a clutch hub 62 associated with transfer clutch 46. PTU 26 is a right-angled drive mechanism including a ring gear 254 fixed for rotation with drum 60 of transfer clutch 46 and which is meshed with a pinion gear 256 fixed for rotation with propshaft 28. According to the present invention, the components schematically shown for torque transfer mechanism 238 are understood to be similar to those previously described. In operation, the power transfer system permits drive torque to be adaptively transferred from the primary (i.e., front) driveline to the secondary (i.e., rear) driveline.

Referring to FIG. 13, a modified version of the power transmission device shown in FIG. 12 now includes a second torque coupling 238A that is arranged to provide a limited slip feature in association with primary differential 25. As before, torque coupling 238 provides on-demand transfer of drive torque from the primary driveline to the secondary driveline. In addition, second torque coupling 238A now provides on-demand torque biasing (side-to-side) between axleshafts 24 of primary driveline 14.

FIG. 14 illustrates another modified version of FIG. 12 wherein an on-demand four-wheel drive system is shown based on a rear-wheel drive motor vehicle that is arranged to normally deliver drive torque to rear wheels 32 while selectively transmitting drive torque to front wheels 22 through a torque coupling 238. In this arrangement, drive torque is transmitted directly from transmission output shaft 240 to power transfer unit 26 via a drive shaft 260 which interconnects input gear 244 to ring gear 254. To provide drive torque to front wheels 22, torque coupling 238 is

shown operably disposed between drive shaft 260 and transfer shaft 252. In particular, transfer clutch 46 is arranged such that drum 60 is driven with ring gear 254 by drive shaft 260. As such, clutch actuator 48 can be adaptively actuated to transfer drive torque from drum 60 through clutch pack 64 to hub 62 which, in turn, drives carrier 246 of front differential unit 25 via transfer shaft 252.

In addition to the on-demand four-wheel drive systems shown previously, the power transmission technology of the present invention can likewise be used in full-time four-wheel drive systems to adaptively bias the torque distribution transmitted by a center or "interaxle" differential unit to the front and rear drivelines. For example, FIG. 15 schematically illustrates a full-time four-wheel drive system which is generally similar to the on-demand four-wheel drive system shown in FIG. 14 with the exception that an interaxle differential unit 270 is now operably installed between carrier 246 of front differential unit 25 and transfer shaft 252. In particular, output gear 244 is fixed for rotation with a carrier 272 of interaxle differential 270 from which pinion gears 274 are rotatably supported. A first side gear 276 is meshed with pinion gears 274 and is fixed for rotation with drive shaft 260 so as to be drivingly interconnected to the rear driveline through power transfer unit 26. Likewise, a second side gear 278 is meshed with pinion gears 274 and is fixed for rotation with carrier 246 of front differential unit 25 so as to be drivingly interconnected to the front driveline. Torque transfer mechanism 238 is now shown to be operably disposed between side gears 276 and 278. Torque transfer mechanism 238 is operably arranged between the driven outputs of interaxle differential 270 for providing an adaptive torque biasing and slip limiting function.

Referring now to FIG. 16, a drivetrain for a four-wheel drive vehicle is shown to include engine 18, a multi-speed transmission 20' for delivering drive torque to a primary or rear driveline 16' through a power transmission device, thereafter referred to as transfer case 300. As seen, transfer case 300 has a rear output shaft 302 interconnected between the output of transmission 20' and a rear propshaft 28'. Further, propshaft 28' is shown to drive a pinion shaft 44' for driving a drive axle assembly which, in this example, is similar to rear axle assembly 34A of FIG. 9. A secondary or front driveline 14' includes a front propshaft 304 interconnecting a front output shaft 306 of transfer case 300 to a conventional front axle assembly 308. A transfer assembly associated with transfer case 300 includes a first sprocket 310 rotatably supported on rear output shaft 302, a second sprocket 312 fixed to front output shaft 306, and a chain 314 enmeshed therebetween. Transfer case 300 is shown to include a torque coupling 38 for providing on-demand transfer of drive torque from rear output shaft 302 through the transfer assembly to first output shaft 306. As seen, transfer case 300 has an electrohydraulic control system 50' that is controlled by ECU 56 in coordination with electrohydraulic control system 50 associated with torque coupling 38 in drive axle 34A.

Referring now to FIG. 17, a full-time 4 WD system is shown to include a transfer case 300' which is generally similar to transfer case 300 of FIG. 16 except that an interaxle differential 320 is provided between an input shaft 322 and output shafts 302 and 306. As is conventional, input shaft 322 is driven by the output of transmission 20'. Differential 320 includes an input defined as a planet carrier 324, a first output defined as a first sun gear 326, a second output defined as a second sun gear 328, and a gearset for permitting speed differentiation between first and second sun gears 326 and 328. The gearset includes a plurality of

meshed pairs of first planet gears 330 and second pinions 332 which are rotatably supported by carrier 324. First planet gears 330 are shown to mesh with first sun gear 326 while second planet gears 332 are meshed with second sun gear 328. First sun gear 326 is fixed for rotation with rear output shaft 302 so as to transmit drive torque to rear driveline 16'. To transmit drive torque to front driveline 14', second sun gear 328 is coupled to the transfer assembly which again includes first sprocket 310 rotatably supported on rear output shaft 302, second sprocket 312 fixed to front output shaft 306, and power chain 314.

While the electrohydraulic control system 50 has been described and illustrated thus far as having a clutch actuator 48 with an actuator ring 102 that is subjected to uniformly increasing fluid pressures to effect movement of the actuator ring 102 in a rotational direction that tends to increase the compressive clutch engagement force that is exerted on the clutch pack 64, it will be appreciated that the invention, in its broadest aspects, may be constructed somewhat differently. For example, the rotary operator 92a may be constructed as shown in FIGS. 18 and 19. In this example, the electrohydraulic control system 50a can be configured so that the torque that is produced by the clutch actuator 48a can be varied based on the position of the actuator ring 102a relative to the reaction ring 100a.

With reference to FIGS. 19 through 21, the reaction ring 100a can be generally similar to the reaction ring 100 that is illustrated in FIG. 4. The projecting lugs 112a can be configured to sealingly engage an outer portion 1000 of the hub 1002 of the actuator ring 102a. In the particular example provided, a seal member 1004 is housed in each projecting lug 112a and is configured to form a seal between the reaction ring 100a and the actuator ring 102a as will be described in detail, below. Optionally, one or more hydraulic valves 1008 can be employed to cause the hydraulic pressure that acts on one or more groups of the actuation chambers 132a to be phased relative to other groups of the actuation chambers 132a according to the relative position of the reaction ring 100a and the actuator ring 102a.

The hydraulic valves 1008 can include an associated one of the supply ports 144a and a fluid distribution feature 1010 that will be discussed in greater detail below. One or more of the ports 144a that provide pressurized fluid to the actuation chambers 132a can be spaced or phased so as to provide pressurized fluid an associated one of the actuation chambers 132a when the actuator ring 102a has been rotated to a predetermined location relative to the reaction ring 100a. For example, the ports 144a-1 through 144a-3 are disposed between respective lugs 112a and separator plates 118a (e.g., port 144a-1 is disposed between lug 112a-1 and separator plate 118a-1) when the clutch actuator 48a outputs a minimum compressive clutch engagement force onto the clutch pack 64a and each of the remaining ports 144a are spaced progressively farther from this relative position or datum. In the particular example provided:

the port 144a-4 is spaced by an additional first predetermined offset interval 1020-4, such as 4°, relative to a predetermined datum, such as datum 1022-4, to provide pressurized fluid to the activation chamber 132a-4 after the actuator ring 102a has rotated through an angle that is about equal to the first predetermined offset interval; the port 144a-5 is spaced by an additional second predetermined offset interval 1020-5, such as 8°, relative to a predetermined datum, such as datum 1022-5, to provide pressurized fluid to the activation chamber

132a-5 after the actuator ring 102a has rotated through an angle that is about equal to the second predetermined offset interval;

the port 144a-6 is spaced by an additional third predetermined offset interval 1020-6 relative to a predetermined datum, such as datum 1022-6, to provide pressurized fluid to the activation chamber 132a-6 after the actuator ring 102a has rotated through an angle that is about equal to the third predetermined offset interval; and

the port 144a-7 is spaced by an additional fourth predetermined offset interval 1020-7 (relative to a predetermined datum, such as datum 1022-7), such as 16°, to provide pressurized fluid to the activation chamber 132a-7 after the actuator ring 102a has rotated through an angle that is about equal to the fourth predetermined offset interval.

The actuator ring 102a can be generally similar to that which is described above in conjunction with the embodiment of FIG. 4, except that the separator plates 118a can be integrally formed with the remainder of the actuator ring 102a and the fluid distribution feature 1010, such as a groove, can be formed in the hub 1002 of the actuator ring 102a, and one or more of the separator plates 118a-4 through 118a-7 can be configured to selectively direct pressurized fluid from an associated supply port 144a to an associated return port 150a (e.g., supply port 144a-4 can selectively supply fluid pressure directly to return port 150a-4).

Each fluid distribution feature 1010 can be configured to permit fluid communication between corresponding supply ports 144a and return ports 150a when the actuator ring 102a has not rotated relative to the reaction ring 100a by a corresponding actuating interval. The actuating interval may be any appropriate interval but in the example provided, each actuating interval corresponds to an associated one of the predetermined offset intervals (i.e., the actuating interval associated with the fluid distribution feature 1010-4 can extend over approximately 4° of the circumference of the hub 1002 of the actuator ring 102a, the actuating interval associated with the fluid distribution feature 1010-5 can extend over approximately 8° of the circumference of the hub 1002 of the actuator ring 102a, the actuating interval associated with the fluid distribution feature 1010-6 can extend over approximately 12° of the circumference of the hub 1002 of the actuator ring 102a and the actuating interval associated with the fluid distribution feature 1010-7 can extend over approximately 16° of the circumference of the hub 1002 of the actuator ring 102a in the example provided).

Accordingly, when the clutch actuator 48a (FIG. 18) outputs a minimum compressive clutch engagement force onto the clutch pack 64a, the reaction ring 100a and the actuator ring 102a can be oriented as shown in FIG. 20. In this condition, fluid exiting the supply ports 144a-4 through 144a-7 is directed into return chambers 134a-4 through 134a-7, respectively, and as such, the fluid does not exert a force onto the actuator ring 102a that would cause the actuator ring 102a to rotate in an actuating direction (i.e., a rotational direction that tends to increase the compressive clutch engagement force that is output by the clutch actuator 48a onto the clutch pack 64a). Consequently, only the pressurized fluid in actuation chambers 132a-1 through 132a-3 is employed to cause the actuator ring 102a to rotate in the actuating direction. Moreover, the fluid distribution features 1010 permit fluid communication between the actuation chambers 132a-4 through 132a-7 and return chambers 134a-3 through 134a-6, respectively, so as to selec-

tively permit relatively low-pressure fluid to enter or exit the actuation chambers 132a-4 through 132a-7 as the actuator ring 102a is rotated.

When the actuator ring 102a has rotated through an angle that exceeds the actuating angle associated with one of the fluid distribution features 1010, the seal member 1004 that is housed in a corresponding one of the projecting lugs 112a can sealingly engage the hub 1002 of actuator ring 102a and thereby inhibit fluid communication between corresponding actuation chambers and return chambers (e.g., actuation chamber 132a-4 and return chamber 134a-3). Simultaneously, a corresponding separator plate (e.g., separator plate 118a-4) can be positioned relative to the supply port (e.g., supply port 144a-4) so that pressurized fluid is directed into a corresponding actuation chamber (e.g., actuation chamber 132a-4) rather than to the corresponding return chamber (e.g., return chamber 134a-4). In this way, the amount of fluid pressure that acts on the actuator ring 102a can be varied based upon the rotational position of the actuator ring 102a relative to the reaction ring 100a.

Configuration of the clutch actuator 48a in this manner can be advantageous in that the torsional output of the clutch actuator 48a may be tailored in a desired manner without significant changes in one or more of the flow characteristics (e.g., the mass flow rate, velocity, pressure) of the pressurized fluid.

While the invention has been described in the specification and illustrated in the drawings with reference to various embodiments, it will be understood by those of ordinary skill in the art that various changes may be made and equivalents may be substituted for elements thereof without departing from the scope of the invention as defined in the claims. Furthermore, the mixing and matching of features, elements and/or functions between various embodiments is expressly contemplated herein so that one of ordinary skill in the art would appreciate from this disclosure that features, elements and/or functions of one embodiment may be incorporated into another embodiment as appropriate, unless described otherwise, above. Moreover, many modifications may be made to adapt a particular situation or material to the teachings of the invention without departing from the essential scope thereof. Therefore, it is intended that the invention not be limited to the particular embodiment illustrated by the drawings and described in the specification as the best mode presently contemplated for carrying out this invention, but that the invention will include any embodiments falling within the foregoing description and the appended claims.

What is claimed is:

1. A power transmission device comprising:
 - a rotary input member adapted to receive drive torque from a source of drive torque;
 - a rotary output member adapted to transmit drive torque to an output device;
 - a torque transmission mechanism operable for transferring drive torque from said input member to said output member, said torque transmission mechanism including a friction clutch operably disposed between said input member and said output member and an electrohydraulic control system for controlling engagement of said friction clutch, said electrohydraulic control system including a clutch actuator, a fluid pump, a motor driving said fluid pump, and a controller, said clutch actuator including a rotary operator and a thrust mechanism, said rotary operator having first and second components with a plurality of actuation chambers and a plurality of return chambers, said first component being fixed for rotation with one of said input and

output members and said second component adapted to rotate relative to said first component, and said thrust mechanism being operable for applying a clutch engagement force on said friction clutch in response to rotation of said second component relative to said first component, said controller being operable for controlling the fluid pressure supplied to at least one of said actuation chambers and said return chambers to thereby control rotation of said second component relative to said first component of said rotary operator;

wherein said electrohydraulic control system is configured so that said actuation chambers are divided into at least two groups, wherein a first group is configured such that said actuation chamber or actuation chambers included therein contain a relatively high pressure fluid when said second component is rotated relative to said first component in an actuating direction regardless of a relative position of said first and second components and wherein a second group is configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said predetermined actuating interval.

2. The power transmission device of claim 1, wherein each actuation chamber of said second group is coupled in fluid connection to a corresponding return chamber via a fluid conduit when said second component is positioned relative to said first component within said predetermined actuating interval.

3. The power transmission device of claim 2, wherein said fluid conduit is formed between said first and second components.

4. The power transmission device of claim 3, wherein said fluid conduit includes a passage that is formed in one of said first and second components.

5. The power transmission device of claim 4, wherein a seal member is carried by said other one of said first and second components, said seal member being sealingly engaged to said one of said first and second components and inhibiting fluid communication between said actuation chamber of said second group and said corresponding return chamber when said second component is positioned relative to said first component by an amount that exceeds said predetermined actuating interval.

6. The power transmission device of claim 1, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said second group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

7. The power transmission device of claim 6, wherein the predetermined offset interval is about equal to the predetermined actuating interval.

8. The power transmission device of claim 1, wherein the electrohydraulic control system is configured to include a third group of actuation chambers, the third group is configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a second predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first

component in said actuation direction by an amount that exceeds said second predetermined actuating interval.

9. The power transmission device of claim 8, wherein the second predetermined actuating interval is greater than the first predetermined actuating interval.

10. The power transmission device of claim 8, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said third group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

11. The power transmission device of claim 10, wherein the predetermined offset interval is about equal to the second predetermined actuating interval.

12. The power transmission device of claim 8, wherein the electrohydraulic control system is configured to include a fourth group of actuation chambers, the fourth group being configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a third predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said third predetermined actuating interval.

13. The power transmission device of claim 12, wherein the third predetermined actuating interval is greater than the second predetermined actuating interval.

14. The power transmission device of claim 12, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said fourth group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

15. The power transmission device of claim 14, wherein the predetermined offset interval is about equal to the third predetermined actuating interval.

16. The power transmission device of claim 12, wherein the electrohydraulic control system is configured to include a fifth group of actuation chambers, the fifth group being configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a fourth predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said fourth predetermined actuating interval.

17. The power transmission device of claim 16, wherein the fourth predetermined actuating interval is greater than the third predetermined actuating interval.

18. The power transmission device of claim 16, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said fifth group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

19. The power transmission device of claim 18, wherein the predetermined offset interval is about equal to the fourth predetermined actuating interval.

20. The power transmission device of claim 1, wherein said first component of said rotary operator is a reaction ring having a cylindrical body segment and a plurality of radially extending first lugs which define a series of channels therebetween, wherein said second component is an actuator ring having a cylindrical body segment and a plurality of radially extending second lugs which extend into said chan-

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nels so as to define a plurality of said actuation chambers and return chambers, wherein said actuation chambers are in fluid communication with an outlet of said control valve, and wherein said fluid pump is operable to draw fluid from a fluid source and deliver high pressure fluid to said control valve such that selective control of said control valve results in rotary movement of said actuator ring relative to said reaction ring.

21. The power transfer device of claim **20**, wherein said actuator ring is fixed to a drive component of said thrust mechanism such that rotation of said drive component results in translational movement of a driven component of said thrust mechanism for exerting said clutch engagement force on said friction clutch.

22. The power transfer device of claim **21**, wherein said thrust mechanism is a ball ramp unit with a first cam member as its drive component, a second cam member as its driven component, and rollers retained in cam tracks formed between said first and second cam member, and wherein said cam tracks are configured to cause translational movement of said second cam member in response to rotary movement of said first cam member for applying said clutch engagement force to said friction clutch.

23. A power transfer device for use in a motor vehicle having a powertrain and first and second drivelines, comprising:

a first shaft driven by the powertrain and adapted for connection to the first driveline;

a second shaft adapted for connection to the second driveline;

a torque transmission mechanism for transferring drive torque from said first shaft to said second shaft, said torque transmission mechanism including a friction clutch operably disposed between said first shaft and said second shaft, a clutch actuator for engaging said friction clutch and a control system for operating said clutch actuator, said clutch actuator including a fluid pump, a rotary operator and a thrust mechanism, said rotary operator including first and second components which define a plurality of actuation chambers and a plurality of return chamber that are adapted to receive pressurized fluid from said pump, said first component being fixed for rotation with one of said first and second shafts and said second component adapted to rotate relative to said first component in response to the fluid pressure in said actuation and return chambers, and said thrust mechanism is operable for applying a clutch engagement force to said friction clutch in response to rotation of said second component relative to said first component, said control system including a motor driving said pump and a controller for controlling actuation of said motor and regulating the fluid pressure supplied to at least one of said actuation chambers and said return chambers;

wherein said actuation chambers are segregated into a plurality of groups, a first one of the groups being configured such that said actuation chamber or actuation chambers included therein contains a relatively high pressure fluid when said second component is rotated relative to said first component in an actuating direction regardless of a relative position of said first and second components and wherein each of the other groups is configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a corresponding actuating interval and contain said rela-

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tively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said corresponding actuating interval, and wherein each of the corresponding actuating intervals is different so that a quantity of said actuating intervals that are employed to exert a force onto said first and second components to cause said second component to be rotated relative to said first component in said actuation direction is dependent upon said relative position of said first and second components.

24. The power transfer device of claim **23**, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in each of said other groups is shifted relative to said supply ports of said actuation chambers of said first group an offset interval that is unique to an associated one of said other groups.

25. The power transfer device of claim **24**, wherein the offset interval of each of said other groups is about equal to said corresponding actuating interval of said other group.

26. The power transfer device of claim **23**, wherein each actuation chamber of said other groups is coupled in fluid connection via a fluid conduit to a corresponding return chamber when said second component is positioned relative to said first component within a smallest one of said corresponding actuating intervals.

27. The power transfer device of claim **26**, wherein said fluid conduit is formed between said first and second components.

28. The power transfer device of claim **27**, wherein said fluid conduit includes a passage that is formed in one of said first and second components.

29. The power transfer device of claim **23**, wherein said first component of said rotary operator is a reaction ring having a cylindrical body segment and a plurality of radially extending first lugs which define a series of channels therebetween, wherein said second component is an actuator ring having a cylindrical body segment and a plurality of radially extending second lugs which extend into said channels so as to define a plurality of said actuation chambers and return chambers, wherein said actuation chambers are in fluid communication with an outlet of said control valve, and wherein said fluid pump is operable to draw fluid from a fluid source and deliver high pressure fluid to said control valve such that selective control of said control valve results in rotary movement of said actuator ring relative to said reaction ring.

30. The power transfer device of claim **29**, wherein said actuator ring is fixed to a drive component of said thrust mechanism such that rotation of said drive component results in translational movement of a driven component of said thrust mechanism for exerting said clutch engagement force on said friction clutch.

31. The power transfer device of claim **30**, wherein said thrust mechanism is a ball ramp unit with a first cam member as its drive component, a second cam member as its driven component, and rollers retained in cam tracks formed between said first and second cam member, and wherein said cam tracks are configured to cause translational movement of said second cam member in response to rotary movement of said first cam member for applying said clutch engagement force to said friction clutch.

32. A torque transmission mechanism for use in a motor vehicle having a powertrain and a driveline, comprising:
an input member driven by the powertrain;
an output member driving the driveline;

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a clutch pack operably disposed between said input and output members;

an apply plate moveable relative to said clutch pack between a first position and a second position, said apply plate is operable in its first position to apply a minimum clutch engagement force on said clutch pack and said apply plate is operable in its second position to apply a maximum clutch engagement force on said clutch pack;

a clutch actuator for controlling movement of said apply plate between its first and second positions, said clutch actuator including a fluid pump, a rotary operator and a thrust mechanism, said rotary operator having first and second components that are coaxially arranged to define an actuation chamber and a return chamber therebetween which are adapted to receive pressurized fluid from said pump, said first component of said rotary operator is fixed for rotation with one of said input and output members and said second component is adapted to rotate relative to said first component in response to the fluid pressure in said actuation and return chambers, and said thrust mechanism is operable to move said apply plate between its first and second positions in response to rotation of said second component relative to said first component; and

a control system including a controller for regulating the fluid pressure supplied to at least one of said actuation and return chambers;

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wherein said first and second components cooperate to define a plurality of hydraulic valves, each of said hydraulic valves being configured to relieve fluid pressure in an associated one of said actuation chambers when a position of said second component relative to said first component is within an associated actuating interval.

33. The power transmission device of claim **32**, wherein each associated actuating interval is different.

34. The power transmission device of claim **32**, wherein each hydraulic valve includes a hydraulic conduit that is configured to transmit fluid between one of said actuation chambers and an associated one of said return chambers when said position of said second component relative to said first component is within said associated actuating interval.

35. The power transmission device of claim **34**, wherein said fluid conduit includes a groove that is formed in one of said first and second components.

36. The power transmission device of claim **34**, wherein each hydraulic valve includes a supply port and wherein a location of at least two of said supply ports relative to a datum is staggered so that relatively high pressurized fluid is directed into said actuation chambers at different times depending on a position of said second component relative to said first component.

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